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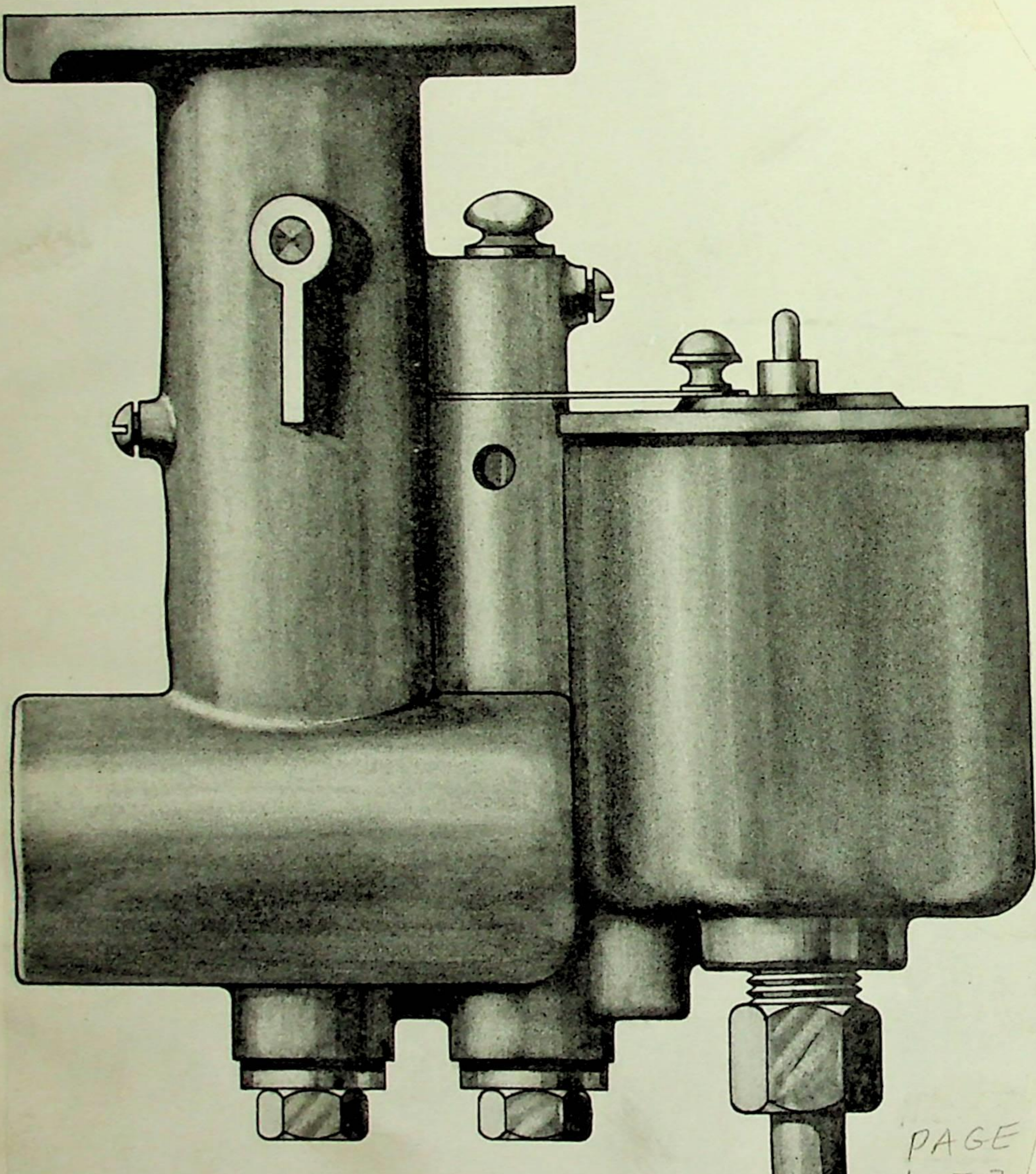

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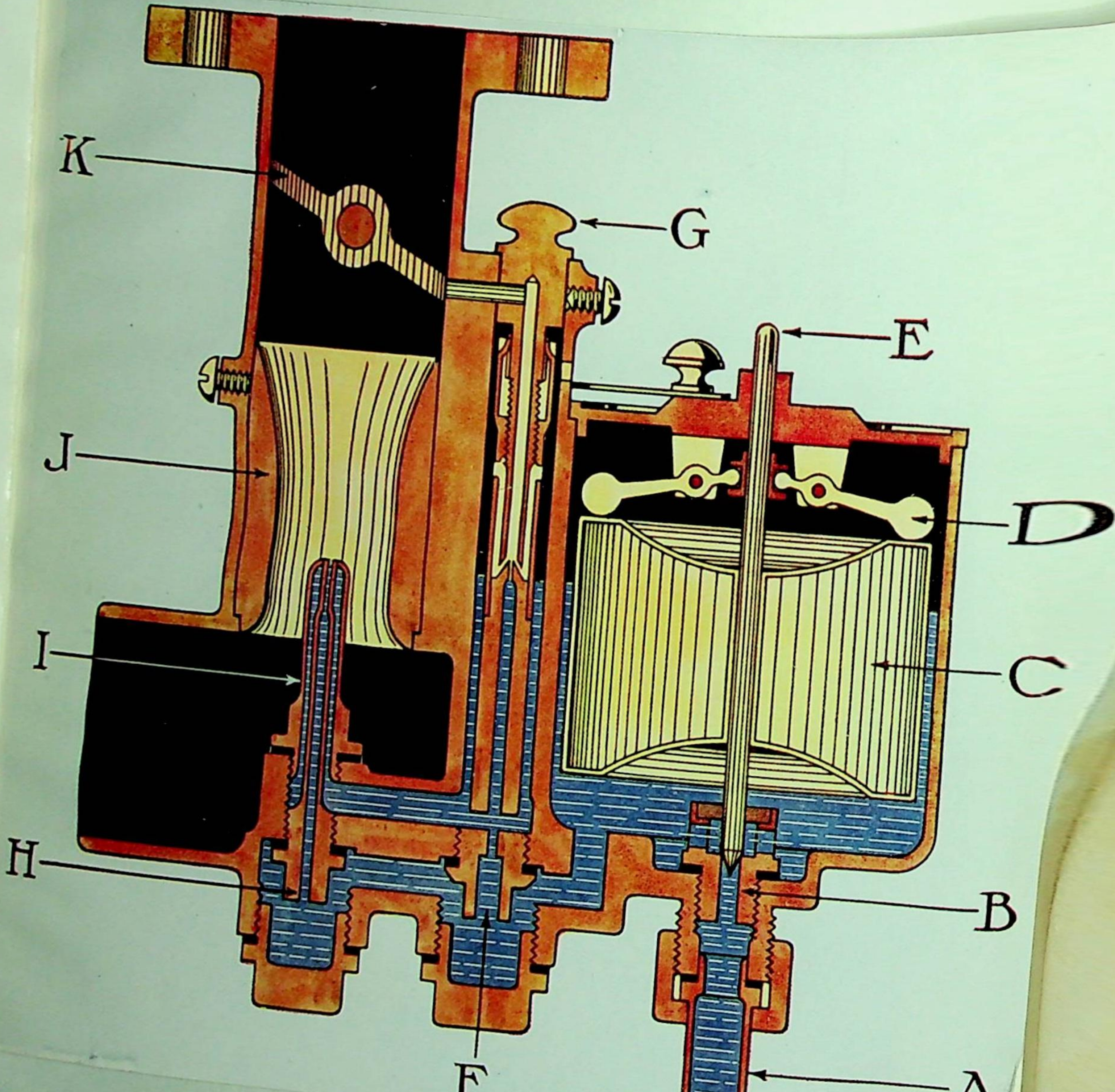
MODEL OF ZENITH CARBURETTOR

A. Petrol Union Nipple.
B. Float Needle Seating.
C. Float.
D. Counterweight.
E. Float Needle.
F. Compensating Jet.
G. Slow-running Device.
H. Main Jet.
I. Main Jet Cover.
J. Choke Tube.
K. Butterfly Throttle.



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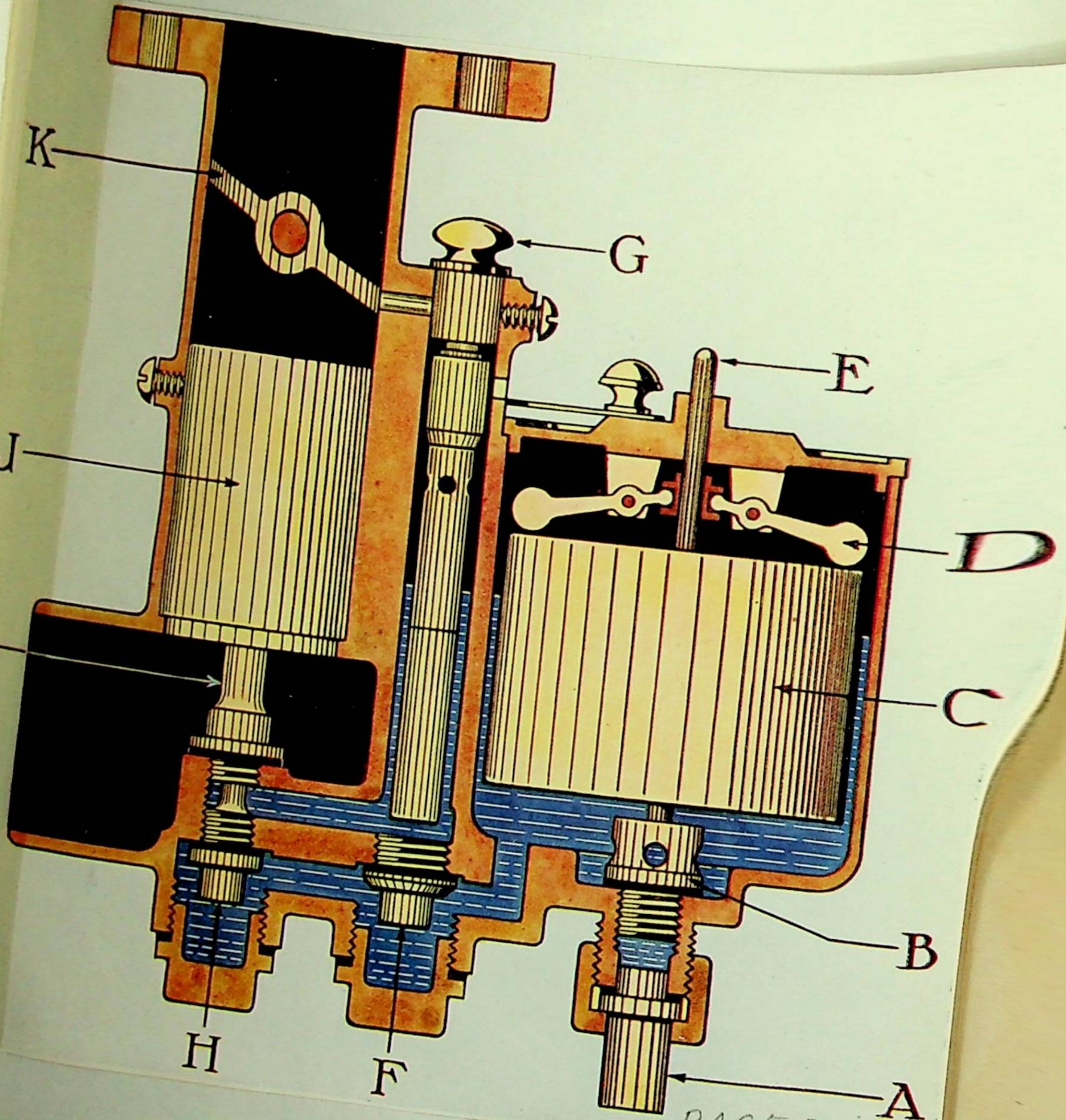

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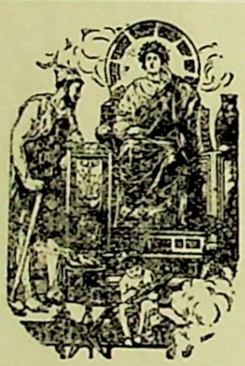

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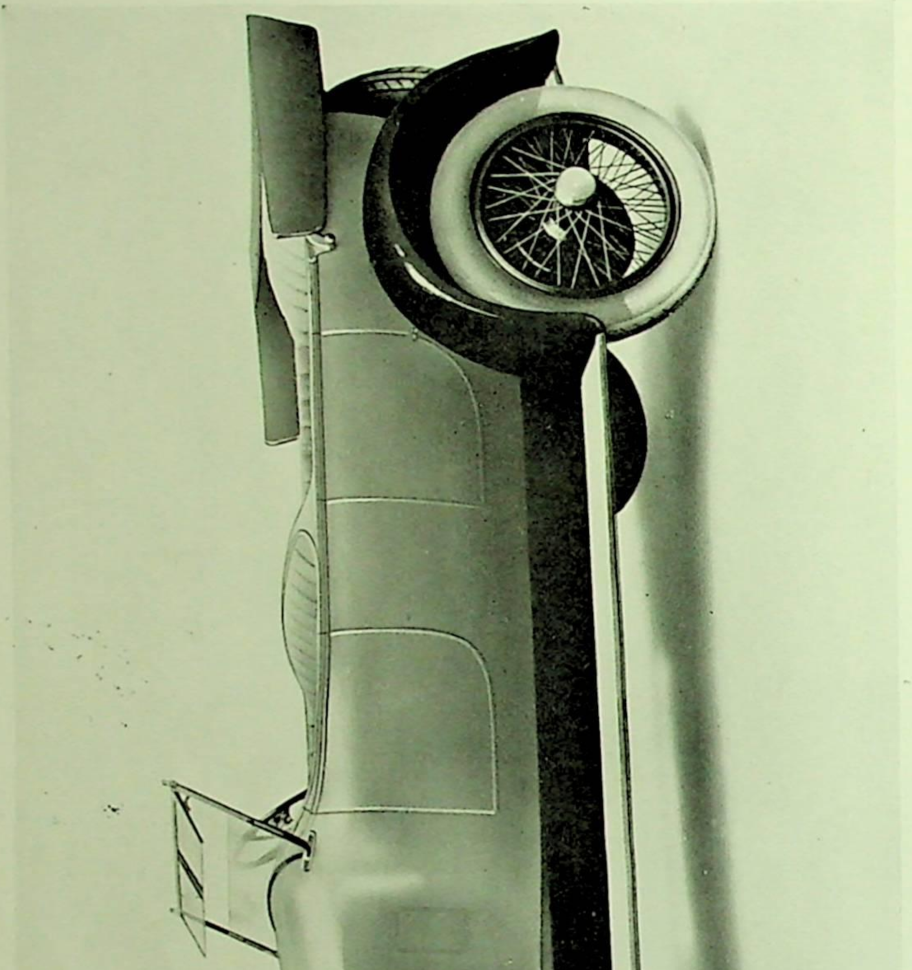
MODERN MOTOR CARS

PLATE I



20 H.P. Rolls-Royce Touring Car.

MODERN MOTOR CARS

THEIR CONSTRUCTION, MAINTENANCE, MANAGEMENT, CARE, DRIVING, AND RUNNING REPAIRS

WITH SPECIAL SECTIONS ON
CYCLE CARS, COMMERCIAL CARS, AND MOTOR CYCLES

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VOLUME I

CAXTON PUBLISHING COMPANY, LIMITED
CLUN HOUSE, SURREY STREET, LONDON, W.C.2

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PREFACE

THE developments in connection with motor vehicles which have taken place during the last ten years have been considerable, with the result that much of the technical literature is out of date and a new presentation of the subject has become necessary.

The aim of this work is to present the more important aspects of the modern motor vehicle in as clear and concise a manner as possible to all those interested, more especially to motor owners, drivers, engineers, students, and garage mechanics.

The highly technical and the purely elementary aspects of the subject have been avoided, generally speaking, but sufficient reference is made to certain design data and formulæ to be of real utility to the designer, draughtsman, and student.

Realising the value of good illustrations and diagrams, no pains have been spared to make the present volumes fully complete in this respect--in fact the proportion of diagrams to printed matter is rather higher than is usual in such books; moreover, as many of these illustrations represent the very latest practice at the moment of writing, their utility will be apparent.

The General Editor has had the loyal co-operation of the Section editors, and in particular would acknowledge the services of the following contributors: Major A. Garrard (Lighting), Captain W. E. Dommett (Transmission, Gear Boxes, Commercial Vehicles, Driving, etc.), and John Phillimore (Care of the Car). It is, unfortunately, not possible here to thank individually the various individuals who have contributed information and illustrations. It is hoped that the references to the sources of information which are made in the pages of these volumes will be accepted as sufficient acknowledgment in these cases. But particular mention may be made of the assistance rendered by *The Autocar*, *The Automobile Engineer*, The Institution of Automobile Engineers, The Daimler Co., The Wolseley Motor Co., The Lanchester Motor Co., Morris Motors, Ltd., Dormans, Ltd., Messrs. Smith & Sons (M.A.), Ltd., Vauxhall Motors, Humber, Ltd., The Austin Co., and Messrs. E. Hoyle and C. Hall.

A. W. JUDGE.

FARNHAM, SURREY.

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MODERN MOTOR CARS

VOL. I

INTRODUCTION

THE modern motor vehicle is of comparatively recent origin. In spite of its high standard of development and its very wide applications, practically the whole of its evolution and progress has occurred within a period of less than forty years, whereas its confrère, the steam-propelled vehicle, dates back some hundred and fifty years or so.

The reason for this more rapid development of the former type of vehicle is not far to seek when it is remembered that not only is it more efficient, but it is considerably lighter, less bulky, and, generally speaking, very much more convenient for the diverse purposes for which it is employed. It is difficult in these days to conjecture what would have been the result had the petrol type of engine remained undiscovered. Possibly we should have developed the gas engine with its gas cylinders or gas bags, so familiar during the period 1914–1918, or should have endeavoured to propel our bicycles, cars, and tanks with very highly developed, compact steam engines.

But the fact remains that from first principles the steam engine cannot compete with the petrol engine on the lighter types of motor vehicles, in the matters of economy, lightness, smallness of bulk, and general convenience. It may be of interest to trace briefly the development of the steam vehicle as applied to road propulsion, and to refer to the later birth and evolution of the internal-combustion engined road cars and cycles.

As early as 1619 a patent was applied for in this country for a "horseless carriage" by Ramsay and Wildgoose, but it was left to a Frenchman named Cugnot to design a vehicle which would actually travel along a road. Cugnot's three-wheeled, steam-propelled car carried two persons at a speed of about two miles an hour along a road in 1770. The success of this experiment appears to have inspired others, for several road vehicles of various designs were produced during the next twenty years, though no startling degree of success has been recorded in connection with these.

To Trevithick, an Englishman, belongs most of the credit for real progress in road locomotion, for during the period 1801 to 1803 he constructed a carriage fitted with a horizontal steam boiler, and driven by means of a steam engine, the crankshaft of which was coupled to the driving or road wheels (which were of 10 feet diameter) by means of gearing. This vehicle attained a speed of ten miles an hour.

Passing over the intervening period of twenty-odd years, during which enthusiastic inventors built walking machines, driven by steam, and road cars so heavy that they could hardly propel their own weight, we reach the year 1825, when W. H. James and Sir J. Anderson made journeys in a steam-propelled car fitted with a tubular steam boiler, travelling with a load of fifteen passengers at twelve miles an hour. Three years later Sir G. Gurney had made such progress that he was able to travel from London to Bath on his steam car, which weighed only a few hundredweight (the modern two-seater car varies in weight between the limits six to twenty hundredweight).

Another pioneer was W. Hancock, who constructed his steam vehicles with boilers working at a steam pressure of 100 lb. per square inch, and achieved much success on account of the quiet running of his road cars.

It is not possible here to dwell on the various contributions to the development of steam vehicles, by men famous in their own times, such as Nasmyth, Dance, Heaton, Ogle, Church and others, but it was mainly due to their foresight and efforts that the steam-propelled vehicle attained its later successes. Considerable opposition, of a legislative nature, was made to the use of mechanically propelled road vehicles, for no fewer than fifty-four Bills were introduced into Parliament on the subject of their taxation during the year 1832, and later during the period 1860 to 1870 the road vehicle was practically swept off the English roads on account of the legislative restrictions imposed. The passing of the Locomotives on Highways Act in 1896 was the cause of the reappearance of the mechanically driven road vehicle. It is only a matter of some thirty years or so since petrol-driven vehicles were compelled to proceed under the guidance of a man who walked in front, carrying a red flag; the maximum speed allowed was four miles per hour.

The steam car has continued to develop to this day, and although in this country practically no new steam passenger cars are now being produced, yet for commercial purposes the steam vehicle still holds a strong position, more especially in the case of heavy cars of from two to five tons.

Reverting to the internal-combustion engined, or motor vehicle, it was due to Gottlieb Daimler, in 1886, that one of the first vehicles, namely, a motor bicycle, using carburetted air in an engine working on the Otto principle, was able to make road journeys. Benz was another pioneer, for in 1885 he was working quite independently of Daimler, and was successful in constructing, and in the trials of, a motor tricycle.

To Messrs. Panhard and Levassor belong the honour of producing the first successful motor car in 1901, since when the progress has been comparatively rapid. In 1904, it is of interest to note, a Panhard car driven by Levassor, and fitted with a 4 h.p. engine, covered a distance of 750 miles in the "Paris to Bordeaux and back" race in just under fifty hours, the average speed being 15 m.p.h.; a modern car could accomplish the same journey in from 10 to 15 hours.

The progress which has resulted in the present-day product has been due largely to commercial incentives, for the wide fields of application, the commercial opportunities, and the rapid transport qualities of the automobile have been quickly recognised and taken advantage of.

Tracing back, for a moment, the steps in the evolution of the modern motor vehicle, the main lines of progress have been due to the adoption of pneumatic tyres, to the improvement of the engine, to careful study of the principles of combustion in the cylinder, the use of better materials, such as the alloy steels, aluminium alloys, and "frictionless" bearing metals, to the multiplication of the number of cylinders, and general speeding up from the earlier 600 to 800 r.p.m. to working speeds of from 2,500 to 3,500 r.p.m., and to improved reliability.

In spite of its improvements, the modern car follows very closely its earlier predecessors in the arrangement of the engine, gear box, brakes, and differential, just as the modern aeroplane is identical in principle with the original Wright machine; the improvements which have occurred have been in details, reliability, efficiency, range, and in accessories for the comfort and convenience of the passengers.

It is difficult to predict the future development of the motor vehicle, but already one perceives its possibilities in connection with military "moving forts," as trackless vehicles capable of travelling at speeds of 30 m.p.h. over rough ground, and of surmounting obstacles in its path. The utility of trackless vehicles of the "caterpillar" type for crossing huge wastes of territory, such as plains and deserts, in which no roadways exist, is already recognised, whilst the amphibious heavy motor, namely the "tank," has proved its possibilities in this and other countries.

It is probable that future development will lie in the direction of lighter and still cheaper cars for the everyday individual; in more automatic and comfortable cars of the normal type (in which the drawbacks of personal control of the engine and gears will be replaced by automatic means); and in the development of the hybrids between the automobile and the aeroplane; also between the aeroplane and the submarine.

CHAPTER I

THEORY OF THE PETROL ENGINE

UNDOUBTEDLY the most important part of the motor vehicle is that which provides the energy to propel it, and it is for this reason that the power unit, whether it be steam, petrol, oil, electric, or gas driven, merits the detailed attention given to it in this and ensuing chapters.

No matter what the type of the engine, the initial source of the energy derived from it can be attributed to the fuel used. In the case of the steam engine, for example, the steam which provides the power by acting upon the piston is generated in the boiler, the source of heat being the combustion or chemical energy of the coal or oil fuel in the furnace. Similarly, in the case of that general class of explosion engine which is termed the internal-combustion engine, and embraces the gas, oil, and petrol types, the chemical action, or the combustion as it is termed, occurs in the cylinder itself, and the heat energy which results from the combination of the oxygen of the air, supplied in the explosive mixture, and the fuel is the source of the mechanical energy obtained from the engine itself.

In the case of most electric vehicles, the electric power required for propelling purposes is derived from a chemical action within the batteries carried, and the original electric energy used for charging the batteries is derived from dynamos or electric generators driven by steam or internal-combustion engines, so that in this case, also, the original source of the energy is derived from the fuel itself. It will thus be seen that in the case of all motor vehicles of the general types defined, the power unit's function is to convert the heat, or combustion energy, of the fuel into mechanical energy for propulsion purposes; the conversion is not, however, a direct one, as will be seen later, but the intermediate changes in the process of deriving energy from the liquid or gaseous fuel, and converting it into mechanical effort at the road wheels, do not influence the original principles upon which the power unit is based.

Before passing on to a consideration of the working of modern internal-combustion engines a brief survey will here be made of the history and development of this type of engine, with a reference also to the steam engine, since the various stages in the progress of these engines afford a much better appreciation of the present position and of the working of modern engines. In regarding these latter, one is apt to overlook the developments which have occurred in the past and have resulted in progress along definite lines ; a knowledge of the historical side is essential in order to understand the reasons for the adoption of modern types, and, moreover, to anticipate the lines along which continued progress will be made in the future.

The earliest mention of any power-deriving device is probably the engine of Hero of Alexandria, who lived two hundred years B.C. It consisted of a type of steam turbine or spherical vessel pivoted on a central axis, and provided with a number of peripheral bent pipes through which steam could escape and thus rotate the vessel about its axis ; it was similar in principle to the lawn-sprinkling device in which the water escapes from a vertically pivoted bent pipe which it rotates in the process ; the principle involved is one of reaction.

No progress is recorded from this period until the seventeenth century, when Giovanni Porta (in 1606) published a book on the subject of pneumatics, in which he depicted the principle of the steam suction pump. In 1629 Branca designed an engine resembling a water wheel to be rotated by the impact of a steam jet on its vanes.

It is not possible, here, to follow each step in the history of the development of the steam engine, if only for the reason of its comparative unimportance in comparison with the development of the explosion type of engine ; but in passing one is tempted to mention the efforts of pioneers such as Thomas Savery, who in 1698 patented a water-raising engine employing high-pressure steam ; to Newcomen, who in 1705 built his famous atmospheric engine for pumping purposes ; and finally to that engineering genius James Watt, an instrument maker of Glasgow, who invented the steam jacket for the cylinders, the condenser for recovering water from the used steam, the double-acting steam engine, beam engine, indicator for showing the manner in which the steam pressure varied in the cylinder, and other improvements, many of which are the fundamental bases of modern steam-engine practice. Considerable progress was due to Trevithick, in 1800.

Passing along to the first application of the steam engine to locomotion, as we have already seen in the introduction to this section, Cugnot was probably the earliest pioneer to actually utilise the steam vehicle, after which attention was given to the problems involved by various inventors in this and other countries.

It would be unfitting to conclude this consideration without a passing reference to the inventor of the first successful steam-driven locomotive, namely, George Stephenson, whose engine, "The Rocket," carried out trials between Stockport and Darlington in 1829, and not only outpaced its competitors, but settled for all time the question of the substitution of the steam for the horse-propelled railway vehicle.

Having outlined briefly the development of the steam engine, let us now turn to the subject of the internal-combustion engine, leaving out a detailed reference to the hot-air engines of Stirling, Rider, Ericsson, and others of the eighteenth century, since time has proved the total unsuitability of this type of engine to any commercial purposes.

The earliest practical example of an internal-combustion type of engine is probably the gunpowder engine of Huyghens, who in 1680 constructed an engine in which a weight was raised by the vacuum formed after the explosion of a gunpowder-air mixture in a closed cylinder fitted with a piston. The principle of this "engine," if it may thus be termed, is shown in Fig. 1. The action of this engine was as follows: the piston being at the top of the cylinder, gunpowder was exploded inside the cylinder, and upon the air, or products of combustion, being driven out, a partial vacuum was formed inside. The atmospheric pressure then forced the piston down into the cylinder and by means of the cord shown raised a weight. Lenoir in 1860, after several inventors had, with varying degrees of success, produced internal-combustion engines employing gas, made the first real practical progress when he constructed a gas engine which worked satisfactorily, rapidly, and silently, and was considered to be a serious rival to the steam engine. Before the year 1865 no less than 400 engines of the Lenoir type were made in France and in this country, but it must not be concluded from this that the engine was in any way perfect, for it is recorded that its gas consumption was very high and that large quantities of cooling water were required for the cylinder jackets.

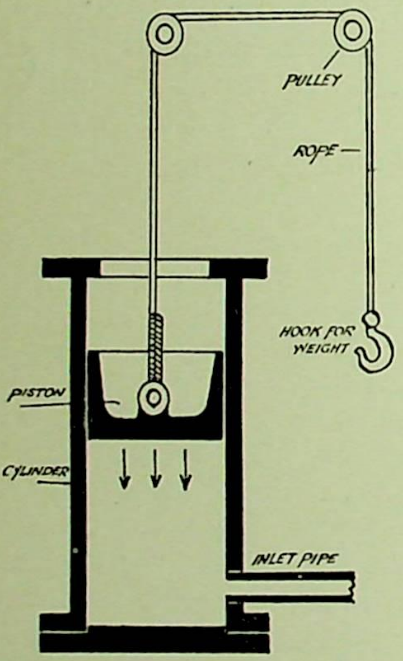

Fig. 1.

The method used by Lenoir in his later engines was to suck into the cylinder a mixture of gas and air, by utilising part of one stroke of the piston. When the piston was half-way along, the mixture was exploded by means of an electric spark provided from an induction coil. The explosion forced the piston along so as to complete the half-finished stroke; the gas was prevented from escaping from the inlet pipe by the fitting of a non-return valve. The movement of the piston was communicated to a crankshaft by the usual connecting-rod method, and as a large flywheel was affixed to the crankshaft, the piston was not stopped at the end of its stroke (when an outlet or exhaust valve was opened, mechanically), but, due to the inertia of the heavy revolving flywheel, it was caused to reciprocate back along the cylinder, thus driving out the products of the explosion. The momentum of the flywheel was sufficient to carry it back again, when a fresh charge of gas and air was drawn in, and when the piston was half-way along another electric spark occurred and the cycle of operations repeated.

Lenoir's earlier engine resembled closely a horizontal steam engine, as will be noticed from the illustration given in Fig. 2, and it was really a converted steam engine in which the slide valve, instead of admitting steam, allowed a mixture of gas and air to enter ; similarly the exhaust valve was of the slide-valve type, as will be seen from Fig. 3. In this illustration the ignition plugs are shown in the two ends of the cylinder. The piston was of the double-acting type as distinct from the single-acting type previously described, and which is the present-day type of petrol engine employed. The troubles experienced with this engine were due to bad lubrication, excessive cylinder temperatures (the piston often became red-hot and fired the incoming charge), and to bad starting, for it will have been observed that the incoming explosive mixture was at the ordinary pressure of the outside atmosphere, and therefore not compressed as the charge of a modern engine is.

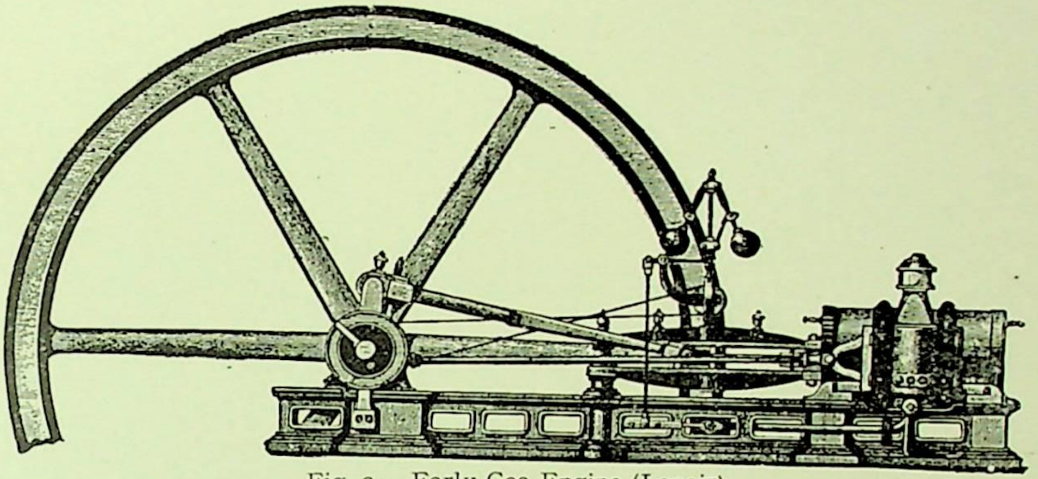
Fig. 2.—Early Gas Engine (Lenoir).

As will be shown later, it is the compression of the charge (to five or six atmospheres pressure) which enables an internal-combustion engine to run economically and to start up without much difficulty; it is easy to understand now why Lenoir's engine, working as it did with little or no compression, consumed about seven times as much fuel as a modern engine of the same horse-power does.

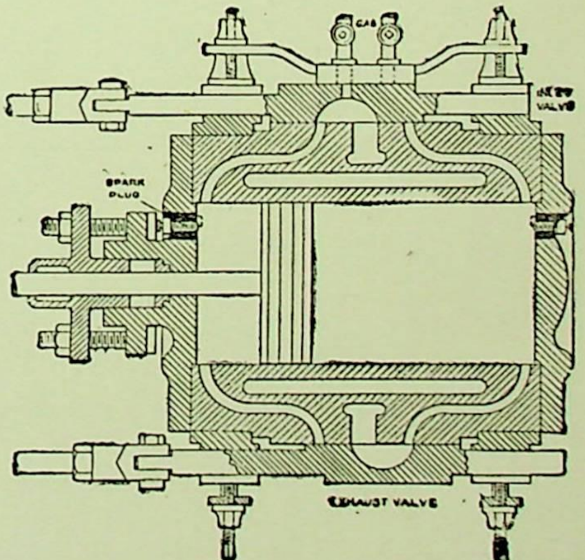

Fig. 3.—Cylinder Section of Lenoir Engine.

Twenty-five years later a single-acting compression type of engine was brought out by Lenoir, and this engine, which represented the fruits of Lenoir's incessant labours and improvements, was certainly a marked advance over the earlier ones; it is interesting to record the fact that this engine consumed only about one-quarter of the coal gas that the earlier engine used, for the same power output. Another pioneer of the gas engine was Beau de Rochas, who in 1862 enunciated the first satisfactory theory of the cycle of operations occurring in the cylinder; he was the first to point out the beneficial effects of high-piston speed, of the initial compression of the charge before explosion, and of the expansion as far as possible of the exploded gases.

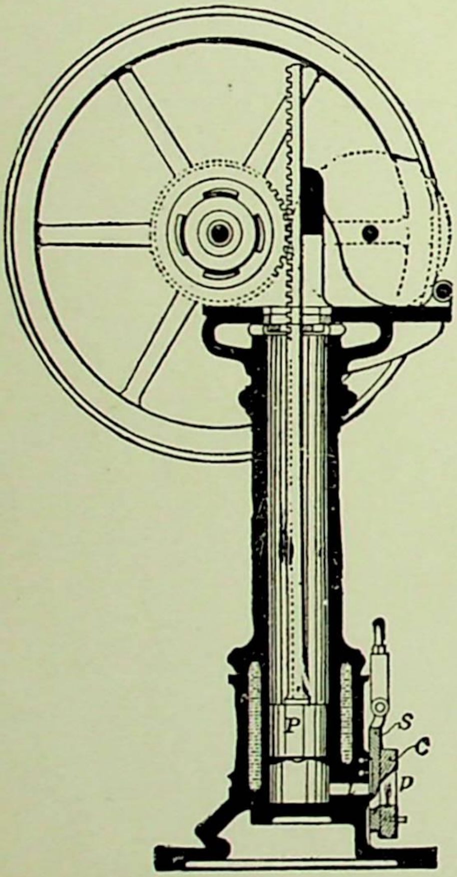

Fig. 4.—Free Piston Otto-Langen Engine.

It is, however, due to a German engineer, Otto, that the foundations of the modern engine were laid. In co-operation with Langen, he produced an engine, in 1866, in which the piston was not connected in the usual way to the crankshaft, but was free to fly vertically outwards on the explosion stroke. When the gases had expanded and cooled sufficiently, the piston's movement was arrested and it commenced to descend under the influence of gravity; by means of a rack-and-pinion ratchet device, as shown in Fig. 4, it communicated its motion to a shaft fitted with a flywheel. On the return or down stroke the piston opened an exhaust or outlet valve of the sliding type, and when the piston had descended to its full amount it was mechanically raised a few inches so as to draw in a fresh charge of gas and air. The mixture was then ignited by means of a gas flame which was arranged in a kind of hollow turn-cock fitted in front of a slot in the cylinder. As soon as the gas-mixture inlet had been shut off, in the cylinder, the turn-cock was rotated so as to expose the mixture in the cylinder to the action of the gas flame.

This engine, interesting although it was, did not work very satisfactorily, as one might imagine after a consideration of spasmodic effects and shocks resulting from a free piston flying outwards in a long cylinder. It is recorded that the Otto and Langen engine consumed from 30 to 50 cubic feet of coal gas per hour for every horse-power given out; this was about one-third of the consumption of Lenoir's earlier engine, and therefore a marked improvement thereon.

In 1876 Otto produced a new engine, which he termed the "silent engine," from the fact of its relative silence in comparison with the early "free-piston" type.

As this engine is of great importance as the forerunner of the modern type, it will be necessary to give it due consideration here; moreover, the reference to it will serve to render clearer the latter explanation of the working of the present-day types of engine. Otto based his engine upon the cycle of operations which Beau de Rochas had defined in his French patent of 1862, in which, as previously mentioned, the advantages of initial compression of the explosive mixture, and subsequent expansion of the explosion products, were emphasised.

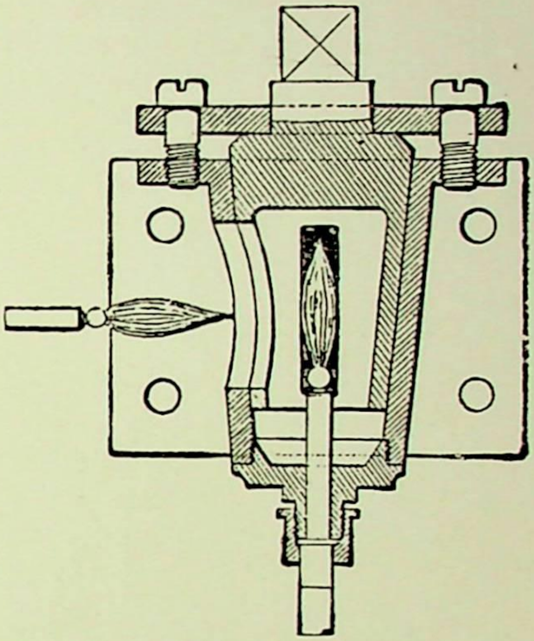

Fig. 5.—Early Flame Ignition.

The cycle of operations proposed by de Rochas and brought to practical realisation by Otto occurs during four consecutive piston strokes, namely, two forward and two backward ones, and occupies two revolutions, during which one explosion or impulse stroke occurs. The diagrammatic illustration shown in Fig. 6 will serve for the purposes of explanation to illustrate the action of the Otto or Rochas cycle.

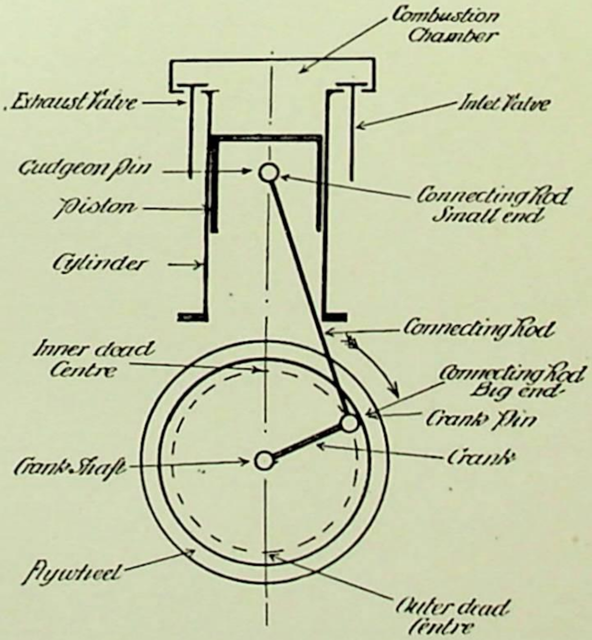

Fig. 6.

The cylinder contains a sliding piston connected to a crank arm of a crankshaft by means of a connecting rod, such that when the piston reciprocates to and fro the connecting rod transmits, and translates, this motion into a rotary movement of the crankshaft. Reference to the diagram of Fig. 6 will also illustrate the various terms generally employed in petrol-engine practice. The sequence of operations, commencing with the piston in the position shown in Fig. 7 (*a*), is as follows, namely:

(1) During the outward stroke of the piston in (*a*), the orifice I is uncovered, due to the covering disc, or *inlet valve*, as it

is termed, being pushed inwards to a small extent. The suction created by the ingoing piston draws into the cylinder a mixture of gas and air, the proportions of which are determined by a special piece of apparatus, termed the *carburettor*, to which reference will be made later.

(2) During the return stroke of the piston, shown in Fig. 7 (*b*), the inlet valve at I is mechanically (or otherwise) closed, and as the mixture cannot escape it is compressed into the valve end, or the *combustion chamber* of the cylinder.

(3) When the mixture is fully compressed, that is to say, when the piston arrives at the outer end of its stroke (away from the crankshaft), a spark or flame is momentarily exposed inside the cylinder, and the mixture at once explodes, driving the piston inwards (towards the crankshaft), as shown in Fig. 7 (*c*). As the hot gases expand, the pressure created forces the piston along to the end of its stroke.

(4) At the end of this stroke (the cylinder being full of hot burnt gases) the *exhaust valve* E is opened and the gases, which are at a pressure considerably greater than that of the outside atmosphere, rush out into the atmosphere or into a silencing vessel, to which reference will be made later. The momentum of the flywheel carries the piston outwards again, sweeping through the exhaust-valve orifice, or *port*, the remaining burnt gases, and the piston is then in the same position again as in (*a*), ready for a new cycle of similar operations. It will be evident that only one side of the piston is utilised, and also that as there is only one working stroke, the momentum given to the crankshaft and flywheel during this stroke must be sufficient to carry the piston over the suction, compression, and exhaust strokes.

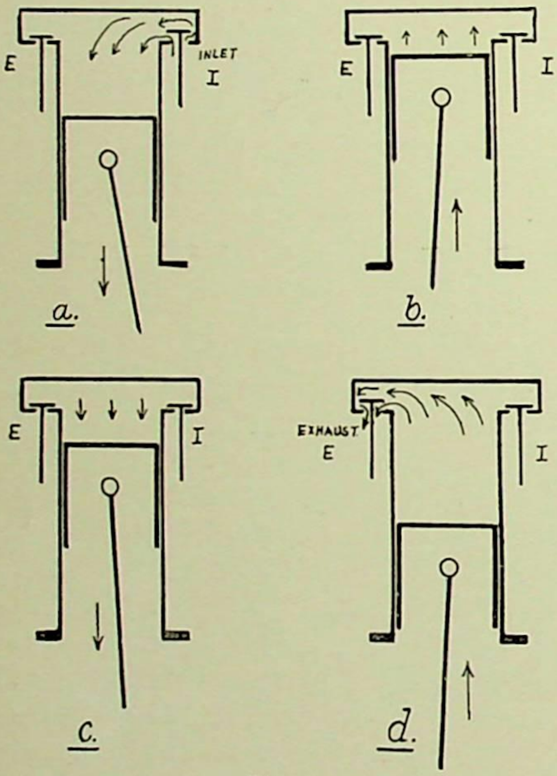

Fig. 7.

In the case of the Otto engine of 1876, which worked upon these principles, the cylinder was horizontal, and the mixture was exploded by means of a gas flame operating in a manner similar to that previously described.

A sectional plan view of the Otto engine is given in Fig. 8, in which the piston, connecting rod, and crankshaft can be clearly seen. The spaces around the cylinder walls and the combustion head were water-jacketed, as will be seen from the illustration, for the purposes of keeping the cylinder sufficiently cool during the working operations ; if water jackets were not provided, the cylinder and piston would become so hot that the incoming mixture would ignite as it was entering and would fire back into the mixing device.

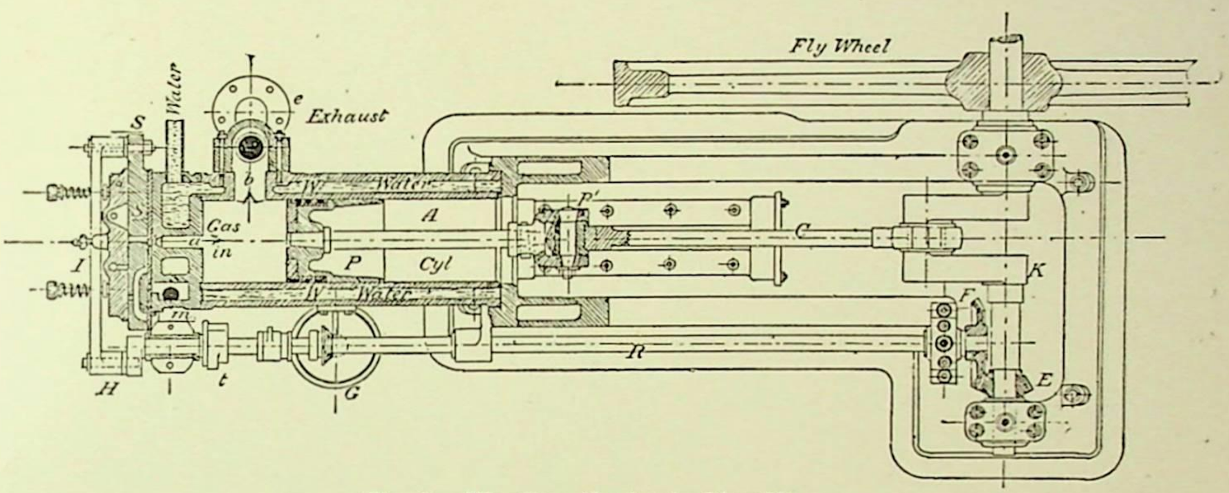

Fig. 8.—The Otto Engine in Plan View.

Otto's engine contained a number of ingenious features which cannot be described here, but mention must be made of the centrifugal type of governor for regulating the amount of gas supplied, and therefore the speed and output of the engine. The oiling device for lubricating the various working parts consisted of a rotating shaft having a crank, or projection, which dipped periodically into an oil-trough and carried off a drop of oil, which was wiped off at the top of the revolution by a fixed pin ; the oil thus wiped off was led down a pipe to the different working parts. During the first ten years following the appearance of Otto's second engine no less than 30,000 engines were built and sold ; the modern gas engine is very similar to Otto's in many respects.

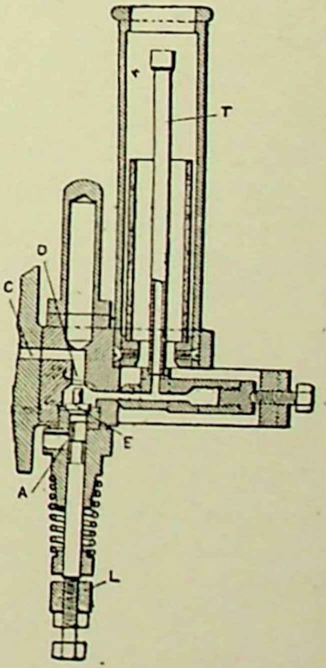

Fig. 9.—Hot-tube Ignition.

Before passing on to a consideration of the petrol or spirit-vapour type of internal-combustion engine, reference will be made to an improved ignition device, known as the *ignition tube,* and which was employed in 1888 on the Otto engine. Fig. 9 shows a section through a typical ignition tube, which superseded the slide-valve type. A tube of iron, T, was heated to and kept at a bright-red heat by means of a Bunsen flame, and by means of a small valve, D, known as a "timing" valve. At the correct moment for the ignition of the charge this valve was opened by means of a cam, L, and the explosive mixture which entered it under the influence of the compression pressure in the cylinder was fired. The iron tube was about three-eighths of an inch in diameter, and was surrounded by a clay or asbestos mantle situated inside a cast-iron chimney. The passage to the engine cylinder is shown at C, the timing valve at E, and the operating lever at L. During the compression stroke the valve E was lifted, thus closing the small end D and opening the tube T to the atmosphere, so that any gases in it could thus escape. At the moment of ignition the cam L allowed the valve E to drop and opened the tube into the cylinder, thus enabling part of the compressed charge to flow into the hot tube, there to be ignited, and in its turn ignite the main charge in the cylinder.

Although this principle is employed on modern gas engines, it has not survived on automobiles, electric ignition being now practically universal.

In the case of certain oil engines using a thick heavy oil, examples of which, namely, the Tartrais, Diesel, and others,[1] are now actually used on motor cars and commercial vehicles, the charge is ignited without any external device, such as sparking plugs or ignition tubes, the principle employed being to maintain a portion of the combustion head at a temperature sufficiently high to ignite the compressed mixture at the right moment. In the case of the Diesel engine only the air is compressed in the cylinder, and the heat generated by this very high compression (in the vicinity of 500 lb. per sq. inch) is sufficient to ignite the fuel-air mixture when the fuel is injected at the end of the compression stroke.

In 1880 Dugald Clerk invented a new type of explosion engine, in which twice the number of explosions were obtained than in the same revolutions of the Otto engine, owing to the fact that each cycle of operations occupied only two working strokes of the piston, instead of the four strokes in the Otto cycle. This type of engine was designated the *two-stroke* or *two-cycle* engine; reference to this type will be made at a later stage in the present chapter, as it is an important one, and likely to become more universal for motor-vehicle use.

Reverting for a while to the early history of the forerunner of the modern petrol type of engine, we find that to Gottlieb Daimler and to Benz, in 1884 and 1885, respectively, belong the credit for producing practical engines working with an explosive mixture of air and benzene vapour or naphtha. In the latter engine, the Otto or *four-stroke cycle* was employed, and a light petroleum spirit or naphtha was employed. The latter was contained in a reservoir which could be heated either by hot water from the cylinder-cooling system or by heat from the exhaust gases. The explosive mixture was obtained by drawing air through the spirit in the reservoir, under the action of the suction of the piston on the inlet stroke. When charged with inflammable vapour, the mixture passed through a safety valve to another admission valve, which enabled it to enter the mixing chamber. In this chamber the richly laden air was diluted with more air, so that the mixture became of suitable proportions for combustion, and this mixture was admitted to the engine through an automatic valve worked by the suction of the engine. The carburetted charge was compressed in the usual way, fired by an ignition tube heated by a naphtha lamp, and expanded until the exhaust valve opened to allow the burnt gases to escape. This engine was regulated in speed by a ball type of governor, of the *hit-or-miss* type, such that when the usual speed was exceeded the governor actuated a lever which closed the admission valve and thus allowed air only to enter the cylinder ; no explosion then occurred until the speed dropped and the governor permitted a fresh charge to enter the cylinder.

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[1] See p. 146.

The Benz engine subsequently fitted to road vehicles had no governor, and utilised electric ignition ; the 3 h.p. model drove a four-passenger carriage.

The engines of Daimler and Benz were of the slow-speed type, the maximum engine revolutions being in the vicinity of 600 to 700 r.p.m., and it was not until 1890 to 1895 that attention was given to the question of the production of more power from a given size of engine, by increasing the speed of the engine. As will be shown later, the power given by an engine increases almost in proportion to its speed, or revolutions, up to a certain limit, so that it was correct in principle for the early petrol-engine designers to attempt to obtain higher power output by increasing the engine revolutions.

Amongst the original exponents of higher-speed engines were Count De Dion and M. Bouton, who were convinced that the future success of the petrol engine lay in obtaining greater power for a given size of engine, and their early engines were capable of a speed of 1,500 r.p.m.

Before passing over this side of the subject, mention should be made of an English inventor, Edward Butler, who designed a three-wheeled vehicle and patented it as early as 1883, although it does not appear to have been built ; the drawings, however, were exhibited at the Inventions Exhibition in 1885. It was in this year that Butler actually built a motor vehicle fitted with a double-acting two-cylinder petrol engine, and employing a magneto for ignition purposes. In 1889 Butler built another improved type of vehicle. This tricycle was driven by a small engine of 2¼-inch bore, of the type mentioned above, and the transmission was through an epicyclic gear, direct to the hub of the rear wheel. A reduction of about six to one was arranged for in the epicyclic gear. It is interesting to note that the steering system was the same as that employed on modern cars, which is known as the Ackerman type.

Further reference will be made to this in a later chapter. Amongst other interesting features of Butler's motor vehicle were its solid rubber-the correct moment for the ignition of the charge this valve was opened by means of a cam, L, and the explosive mixture which entered it under the influence of the compression pressure in the cylinder was fired. The iron tube was about three-eighths of an inch in diameter, and was surrounded by a clay or asbestos mantle situated inside a cast-iron chimney. The passage to the engine cylinder is shown at C, the timing valve at E, and the operating lever at L. During the compression stroke the valve E was lifted, thus closing the small end D and opening the tube T to the atmosphere, so that any gases in it could thus escape. At the moment of ignition the cam L allowed the valve E to drop and opened the tube into the cylinder, thus enabling part of the compressed charge to flow into the hot tube, there to be ignited, and in its turn ignite the main charge in the cylinder.

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Further reference will be made to this in a later chapter. Amongst other interesting features of Butler's motor vehicle were its solid rubbertyred wheels of 32-inch diameter, and its float-feed system of carburation. A speed range of from 4 to 12 miles per hour was obtained on the road, but this maximum speed was unfortunately not possible before the Locomotive on Highways Act in 1896 removed the pre-existing maximum speed limit of 4 m.p.h.

In this brief survey of the development of the power unit of the motor vehicle it should not be overlooked that both the steam and electric vehicles have been through various phases of trial and development during the last twenty-five to thirty years, and that in the earlier period (1895 to 1905) the electric vehicle was certainly amongst the more promising, for it held many speed records. Thus in 1899 Jenatzy attained a speed of nearly 66 m.p.h. on an electrically driven car, and this record was not beaten until 1902, when a steam car designed and driven by Serpollet attained a speed of 75 m.p.h. The impetus thus given to the development of high-speed road cars evidently inspired the petrol-engine adherents of this period, for they were able to construct petrol-driven cars which beat the steam car's record, and established and held the records for speed until 1906, when a steam car, of the Stanley type, attained the amazing speed of 127½ m.p.h. at Ormonde Beach in America. In 1909 a huge-engined racer, driven by a Benz engine, put up this speed record to nearly 141 m.p.h., but it is noteworthy that the steam-engined car has fallen into disfavour since.

The rapid development of the petrol-engine car and the manner in which it has outdistanced its rivals will be evident from the preceding remarks, and the position to-day is that for all types of motor vehicle, from the tiny motor-assisted pedal cycle to commercial lorries and "tanks" employed for military purposes, the petrol-type engine is almost universal.

COMBUSTION AND EXPLOSION

It has already been mentioned that the original source of the energy, or power output, of an internal-combustion engine is a chemical action in which the fuel and the oxygen of the air is concerned.

Let us consider a little more fully how such a chemical action can result in energy being liberated for producing power in a petrol engine, for a complete understanding of the principles is essential to a knowledge of the working conditions in the engine cylinder.

The chemical action with which the present considerations are concerned is that known as *combustion*, that is to say, the chemical combination of a combustible substance, such as petrol, benzole, or paraffin with oxygen.

All organic, or naturally occurring combustible substances, or fuels, are known to be made up of the three principal chemical elements: carbon, hydrogen, and oxygen, in various proportions. These elements are usually defined by the capitals C, H, and O respectively, and the weight-composition of a natural fuel is expressed by a chemical formula which indicates the proportions of the elements present.

For example, the chemical formula for benzene is C_6H_6, which means that this liquid fuel is composed of six parts of carbon and six parts of hydrogen ; the carbon parts, however, weigh considerably more than those of the hydrogen, due to the fact that the atom of carbon is some twelve times heavier than the atom of hydrogen.

When such a fuel is burnt in oxygen, the carbon is converted into carbonic acid gas (or carbon dioxide, expressed by the formula CO_2) and the hydrogen to water (or H_2O). In this conversion of benzene, by the process of combustion, into carbonic acid gas and water, a considerable amount of heat is evolved. It is this heat which is the source of all the power which is supplied to the piston of the engine. The amount of heat which is generated by the combustion of a liquid or gaseous fuel can be gauged by the fact that sufficient heat is evolved when a pint of ordinary petrol is burnt in air to raise 100 pints of water (i.e. 12½ gallons) from ordinary air temperature to the boiling point (100° Centigrade or 212° Fahrenheit).

The type of chemical action with which the internal-combustion engineer is concerned is the combustion of the fuel in the form of a gas or vapour (such as petrol or benzole vapour) with the oxygen of the air, within the confined space of the cylinder, or combustion head of the engine.

When an inflammable vapour is mixed with air (which contains about four parts of an inert or inactive gas, termed nitrogen, and one part of oxygen) in certain proportions, the mixture is found to be explosive, so that when a spark, match, flame, or other similar source of heat is applied, the elements of the mixture combine with more or less suddenness, and the result is termed an explosion.

If petrol vapour and air in certain proportions at the ordinary atmospheric pressure are ignited, the flame travels through the mixture comparatively slowly and no noticeable noise is produced. This low-pressure combustion, which was employed in some of the earlier engines already referred to, and utilised the expansion produced by the heat of the burnt gases to propel or move a piston fitted in the cylinder in which the combustion took place, is known as *inflammation*, and is simply a more or less gentle burning of the explosive mixture from particle to particle, starting from the point of occurrence of the ignition and spreading throughout the mixture, until every particle of the fuel has been burnt.

The process of inflammation is well illustrated in the case of a heap of loosely piled guncotton placed on the ground and ignited with a match ; the mass simply combusts, or burns, until all the combustible substance is used up. Another illustration is that of a mixture of ordinary coal gas and air, in the proportions of about one volume of gas to ten of air, ignited in and from one end of an open long glass tube of about 1 inch internal diameter; the flame can be seen travelling along the tube from the end at which it was ignited to the other open end, the only noise being a gentle "puff."

This type of combustion as it stands is not, however, of much practical utility in connection with internal-combustion engines, for not only is the speed of flame and pressure-wave travel comparatively slow, but the actual pressures which result from the combustion are so small that only very low-power outputs could be obtained from reasonably small engines working under such conditions.

Fortunately, however, there is another well-known physical principle connected with combustion and explosion which can be turned to account, namely, the fact that when an explosive mixture is compressed to some extent before it is ignited, the resulting explosion is considerably quicker and more violent, and the pressures resulting very much higher than in the preceding case.

Another principle which is also closely concerned with internal-combustion engine working is that of *detonation*, in which the explosive mixture explodes with great suddenness and with violence. In this case the combustion is over in a very small fraction of a second. It is not always necessary to compress the explosive mixture beforehand for detonation to occur, for a mixture of coal gas, or petrol vapour, and pure oxygen enclosed in a cylinder can be detonated if suitable precautions are taken.

In order to illustrate the difference between these two extreme types of explosion, for two of the simple gases, consider the case of a mixture of one volume of carbon-monoxide gas and one volume of oxygen enclosed in a vessel. When the mixture is gently ignited, or lit, the flame travels through the vessel at the rate of about 3 feet per second (or about 2 m.p.h.). If, on the other hand, this mixture is violently ignited, as with a percussion cap containing fulminate of mercury, the explosion becomes a detonation, and the flame travels through the mixture at the rate of about 5,000 feet per second (or 3,750 m.p.h.).

To revert to the example previously given of the inflammation of guncotton, if instead of applying a match or flame to this heap of guncotton, it be fired by means of a percussion cap, the explosion will at once become of a detonatory character. In the past, this principle of detonation has been fairly vividly illustrated in the case of the acetylene gas applied to ordinary petrol engines; here the explosion of the acetylene-air mixture compressed in the engine cylinder is sufficiently violent to give a detonation and comparatively high explosion pressures, with the results that several cases have been recorded of the engine cylinders fracturing or blowing off.

Let us consider next how this question of detonation affects the case of the ordinary petrol engine. Evidently, if this detonation were to occur, the action of the engine would be very violent, the pressures very high, and, as a general result, the engine would be inflexible and would refuse to run slowly or quietly. Moreover, the temperature of the explosions would cause the cylinder to become very hot—so hot, in fact, that unless special cooling precautions were taken, the engine would suffer from pre-ignition (that is to say, the gaseous mixture would become ignited before its proper instant, and the resulting pressure generated would act detrimentally against the piston) and the engine would finally refuse to work.

Fortunately, Nature has provided not only the means for detonation, but also the safeguards against it, in providing in the air used for the combustion not only the *oxygen* required, but also a diluting, inert gas, incapable of combustion in the ordinary sense, known as *nitrogen*. The fact that this nitrogen constitutes about four-fifths of atmospheric air, and is inactive in itself, is the explanation of the results which actually occur in normal petrol-engine practice, namely, that the explosions are neither detonatory nor of the inflammation type. If petrol vapour in suitable proportions were mixed with pure oxygen, very violent explosion would occur on ignition, but the effect of the presence of nitrogen in the rather large proportions in which it occurs in the air is to tone down the detonatory nature of the explosion, and to render it capable of being satisfactorily used in the engine, so that a wide range of useful speeds is obtainable.

Having distinguished between the different types of explosion which can occur, we will pass on to a consideration of one or two other interesting properties of explosive mixtures with which the automobile engineer is concerned.

COMPRESSION

We have seen already[1] that the compression of the explosive mixture is an important factor so far as power is concerned, and that as compared with the uncompressed mixture a considerable improvement in pressure and speed of explosion (and incidentally of the engine revolutions) results.

It can be laid down generally that as the petrol-air mixture is compressed more and more, so will the pressure of explosion increase, and also its temperature. The limits of compression are dictated by the facts that if carried beyond a certain limit (in the vicinity of 120 to 130 lb. per sq. inch, as compared with the ordinary atmospheric pressure of 15 lb. per sq. inch), the mixture begins to detonate too readily, the explosion pressures are so high that much stronger and heavier engines must be built, and the question of the adequate cooling of the cylinders and pistons, in order to preclude the eventuality of premature ignition, becomes a serious one. The usual range of working compressions in petrol engines used in automobiles is from about 50 to 90 lb. per sq. inch.

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[1] See p. 7.

One of the most important lines of development of the present-day type of petrol engine has been in the direction of gradually increasing compression pressure, until to-day, with the ordinary fuels (benzole, petrol, and paraffin) employed, the safe upper limits of compression have been reached.

The manner in which the explosion pressure depends upon the initial compression pressure of the explosive mixture will be evident from the following table, in which the compression ratios—that is, the ratios of the volumes of the explosive mixtures before and after compression—are given, together with the equivalent compression pressures, and the approximate explosion pressures which would be obtained in a modern petrol engine working with correctly proportioned mixtures of petrol vapour and air. It is assumed, of course, that pre-ignition is absent.

RELATION BETWEEN COMPRESSION PRESSURE AND EXPLOSION PRESSURE

Compression Ratio.	Compression Pressure in lb. per sq. inch above atmospheric (gauge).	Approximate Explosion Pressure in lb. per sq. inch (gauge).
3·5	66	230
4·0	80	275
4·5	95	335
5·0	111	390
5·5	128	450
6·0[1]	148	520
6·5[1]	166	580
7·0[1]	186	650

A table such as that given is useful to the engine designer to enable him to estimate the stresses which occur as a result of the explosion, and to anticipate the power output of an engine.

RANGE OF EXPLOSIVE MIXTURES IN PETROL ENGINES

It has already been mentioned that there are limits in the proportions of the fuel (to be combusted), and of the air or oxygen supplied for the combustion. If the mixture is too weak in fuel it will not explode, or "*fire*"; and on the other hand, if there is too much fuel present, the mixture also will not ignite, for it becomes choked, as it were. In the latter case the fuel particles so completely overwhelm, or swamp, the oxygen particles (or molecules) that combustion is too slow and the engine fails to work. When too weak in fuel, the excess of nitrogen in the mixture acts in a similar manner as a deterrent to explosive action.

The limits of mixture ratio on which a petrol engine will work vary from the *rich* limit of about 8 parts of air to 1 of petrol (by weight) down to the *weak* limit of about 20 parts of air to 1 part of petrol; the mixture strength which gives the most economical running is one of about 15 parts of air to 1 of petrol, whilst a rather richer mixture of 12 to 14 parts of air to 1 of petrol gives the most power output.

When the explosive mixture has been properly combusted, or burnt, in the engine, the exhaust gases contain only carbonic acid gas (CO_2), nitrogen (N_2), and water vapour. If, however, the mixture is too rich

[1] These values are above the ordinary useful limits.

in petrol, then a mixture of carbon-monoxide (CO) (a poisonous, colourless, and odourless gas, which is present in the *afterdamp* of coal-mine explosions), carbonic acid gas, nitrogen, and water-vapour occurs in the exhaust gases. It is for this reason dangerous to run a petrol engine in a closed shed or garage for long, if it is exhausting into the building itself—several fatal accidents have resulted from this practice. The exhaust gas from a weakly proportioned mixture is characterised by the presence of oxygen as well as the other three constituents of the gases from the correctly proportioned mixture above alluded to.

It will be shown later that the function of a well-designed carburettor is to supply the correct proportions of air and fuel at all engine speeds and loads.

PRINCIPLES OF THE FOUR-STROKE ENGINE

The cycle, or sequence, of operations first laid down by Beau de Rochas in 1862, and carried out by Otto in 1876, is that employed to-day in modern petrol engines of the four-stroke type; a detailed consideration will therefore be given here to the theoretical and practical sides of the subject of this type, before proceeding to a description of the various designs of modern engine working upon this principle.

The Otto cycle has already been described, and we have seen, with the aid of the line-diagrams given in Fig. 7, how the cycle of operations can be applied to an engine. Let us now turn for a moment to the practical application of these principles, and consider how these operations are carried out in the present-day design of petrol engine. For this purpose the diagrams shown in Figs. 10, 11, 12, and 13 have been prepared, and will serve to illustrate not only the names of the various components, but also their functions in the engine itself.

Referring to Fig. 10, which corresponds to Fig. 7 (*a*) on p. 10, it will be seen that there is a cylindrical chamber, termed *the cylinder*, provided with ribs or fins for cooling purposes, and in which another cylindrically shaped part, the *piston*, can reciprocate. The motion of the piston is converted into one of rotation by means of the connecting-rod and crank shown. The piston is made to fit the cylinder very closely, but not to such an extent that it will require too much effort to slide it up and down. Gas-tightness is ensured by fitting two or more piston rings, details of which are given in a later chapter.

The gaseous mixture is obtained from the carburettor (shown on the right) and enters the cylinder through the inlet port, and at the appropriate period through the inlet valve. In the diagram the valve is shown in the open position, so that mixture will flow into the cylinder as the piston moves downwards under the influence of the suction created.

The mechanism for operating the valves is shown in these diagrams, and is similar to that almost universally employed in this type of engine, which is termed a *poppet-valve engine* of the overhead-valve type. It will be observed that there is a gear wheel fitted rigidly to the crankshaft, meshing into two other gear wheels of twice the number of teeth, so that each rotates at one-half the speed of the crankshaft ; the two shafts driven by these larger gear wheels are known as the *camshafts* or the *two-to-one* shafts. These carry the cams, whose function it is to

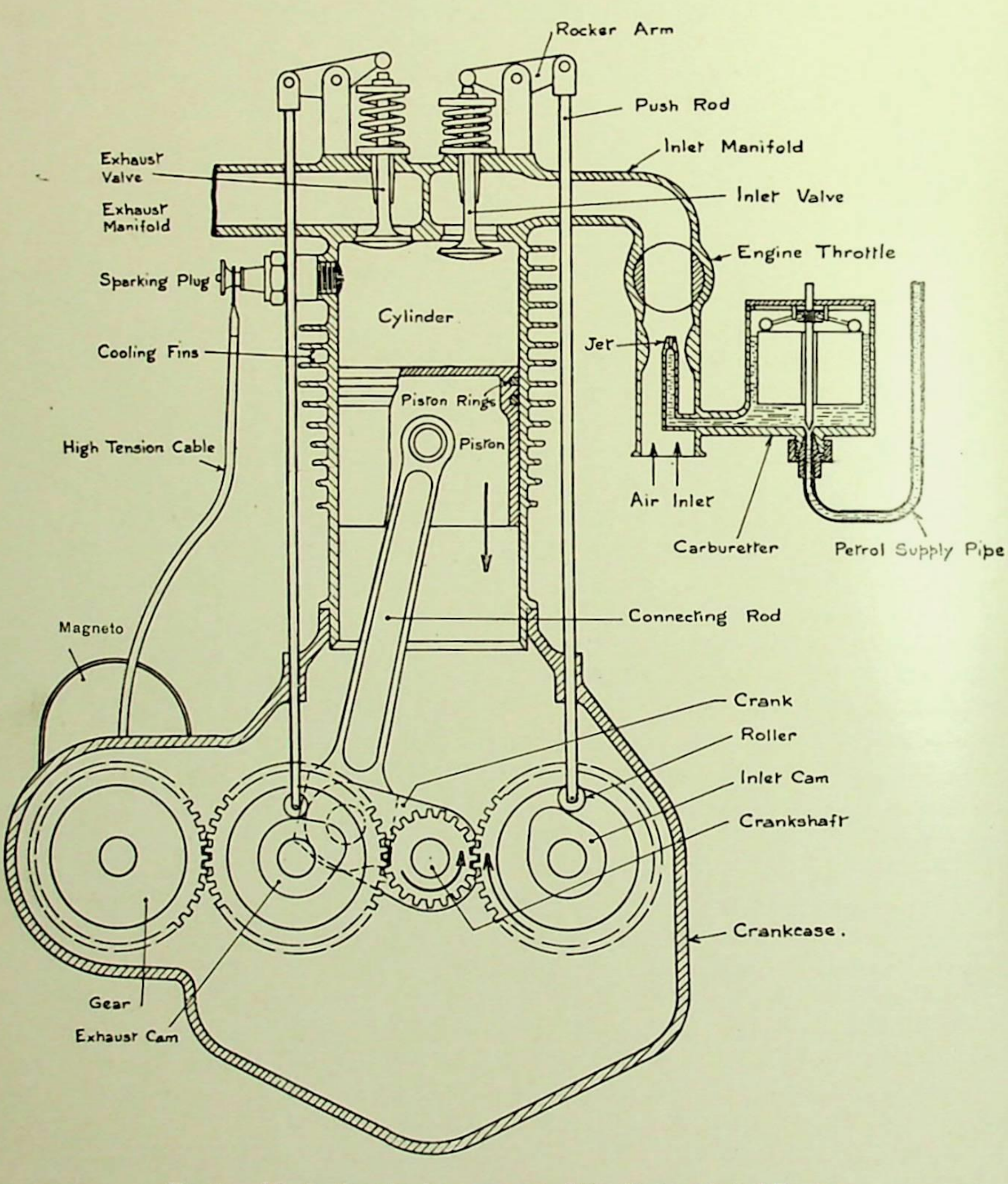

Fig. 10.—Illustrating the Arrangement of the Modern Petrol Engine.

operate the push rods at certain parts of the cycle. The push rods, which are provided with rollers at their lower ends, in turn open the valves through the agency of the rocket arms shown. The coil springs concentrically placed with the valves are for the purpose of closing the valves as quickly as possible when the push rods move downwards again.

The extreme gear wheel shown on the left-hand side of the diagram in Fig. 10 is for the purpose of driving the electrical apparatus, that is, the magneto, which provides the spark at the appropriate moment for firing the compressed charge. The high-tension cable from the magneto to the sparking plug can be seen in the diagram.

Having described briefly the principal components of the petrol engine, we will proceed to a consideration of the working of the engine, referring also to the diagrams shown in Figs. 11, 12, and 13.

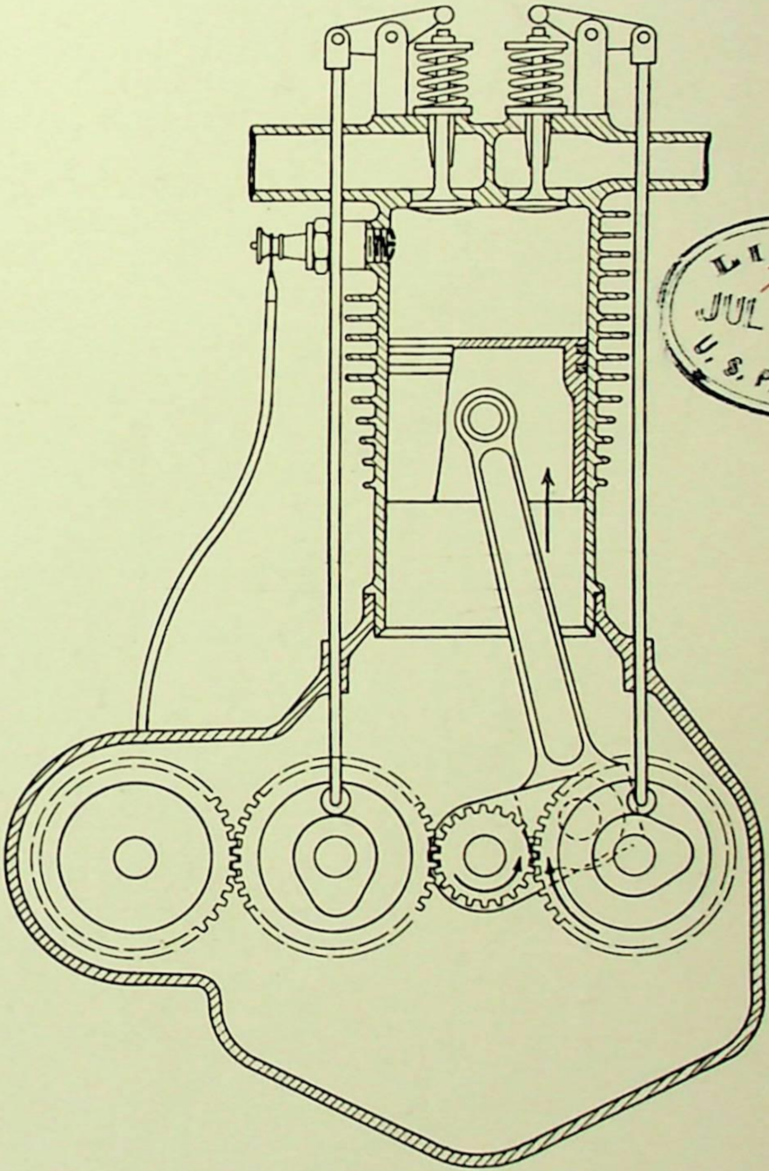

Fig. 11.—The Compression Stroke.

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Commencing with Fig. 10, the piston is moving downwards, owing to the momentum of the flywheel on the crankshaft,[1] which, as has been shown previously, carries the crankshaft around during its idle revolutions under the influence of the effort received during the firing or power stroke. The suction created during this downward stroke draws air through the carburettor in the direction indicated by the arrows on the right, the inlet valve being opened by the cam to allow this to take place. The air in passing through the restricted tube of the carburettor carries petrol with it from the small jet of the petrol supply also shown. The proportions of air and petrol from the carburettor are the correct ones for combustion, and this mixture of air and petrol vapour continues to flow into the cylinder during the whole period of the downward stroke of the piston. When the latter arrives at its lowest position, the inlet valve closes, and the cylinder is then full of a mixture at about the same pressure as that of the atmosphere outside.

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[1] For the sake of clearness, this has been omitted from the diagrams.

On the next upward stroke of the piston, both valves remain closed, and as the mixture cannot escape, it is compressed steadily until the piston arrives at the upper extremity of its stroke. Fig. 11 denotes the compression strokes, the piston being a little less than half-way up.

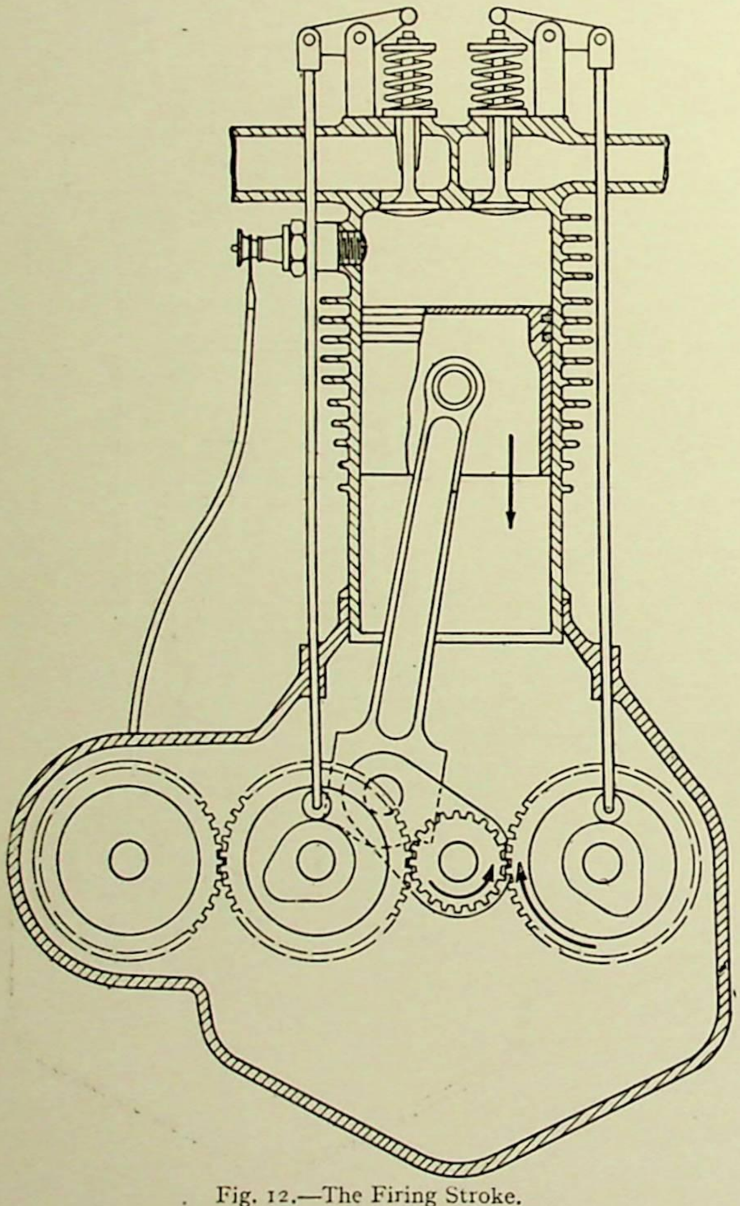


Fig. 12.—The Firing Stroke.

At about the top of the compression stroke, and when the mixture is in a condition of greatest compression in the space above (which is known as the *combustion chamber*), a spark is caused to occur at the sparking plug, and the compressed mixture explodes. In this process of explosion a great amount of heat is evolved which causes the burnt gases to expand considerably, and in expanding to force the piston downwards. It must be clearly understood that it is this expansion of the gases confined in the cylinder which is the working effort from which all the engine power is derived, and that the heat of the combustion of the petrol-vapour and the oxygen of the air is the source of the expansion energy.

Just before the piston arrives at the lower end of its expansion or firing stroke (shown in Fig. 12), the exhaust valve opens and the burnt gases, which are still at a pressure considerably higher than the atmospheric pressure outside, rush out through the exhaust port (usually into a silencing device). This exhaust valve remains open during the whole of the next stroke, as depicted in Fig. 13, and the upward sweep of the piston forces most of the remaining burnt gases out of the cylinder, leaving only the combustion chamber filled with these gases.

The engine has now executed a complete cycle of operations in four consecutive strokes of the piston, that is to say, in two revolutions, and the piston is left in a position ready to repeat this cycle. There is thus one power stroke every two revolutions of the engine, and three idle strokes during which part of the firing-stroke energy is utilised in working the mechanism, in inducing and compressing the explosive mixture, and in ejecting the exhaust gases.

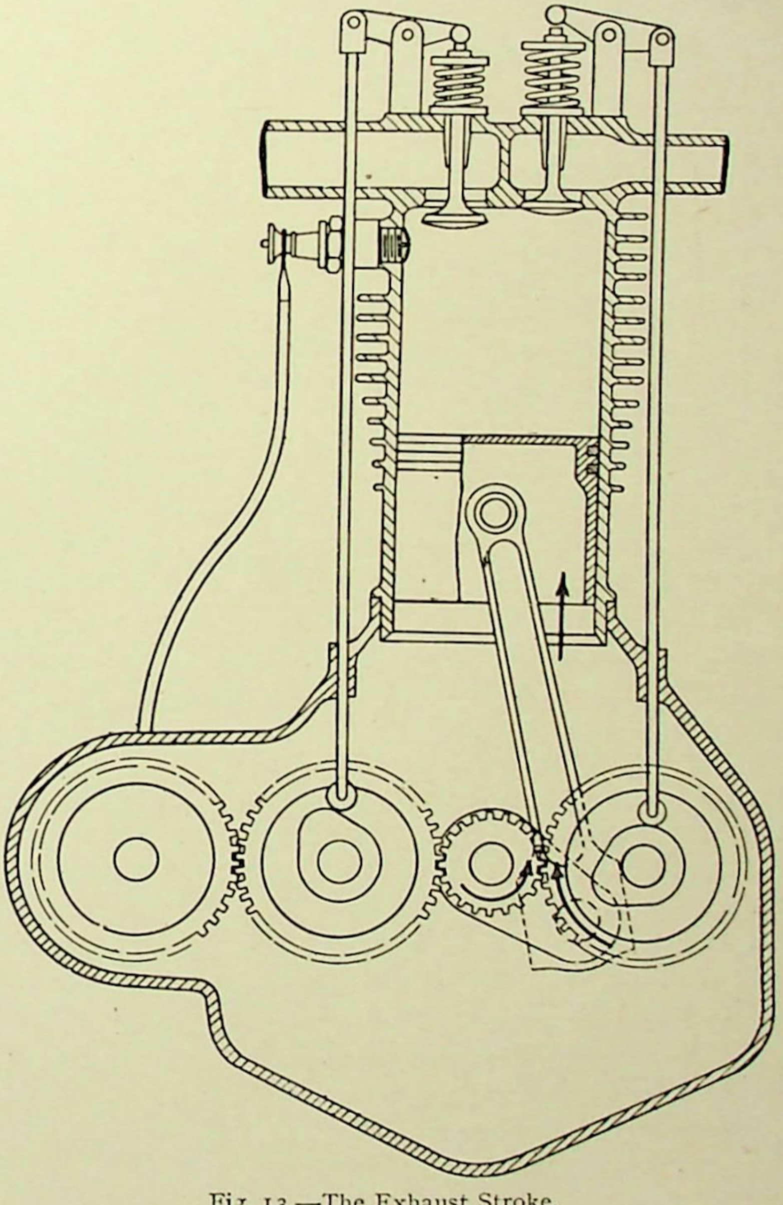


Fig. 13.—The Exhaust Stroke.

PRESSURES IN THE CYLINDER

From what has gone before, it will be observed that there is a continuously changing pressure within the cylinder during a cycle of operations; it is of interest to examine these changes of pressure a little more fully.

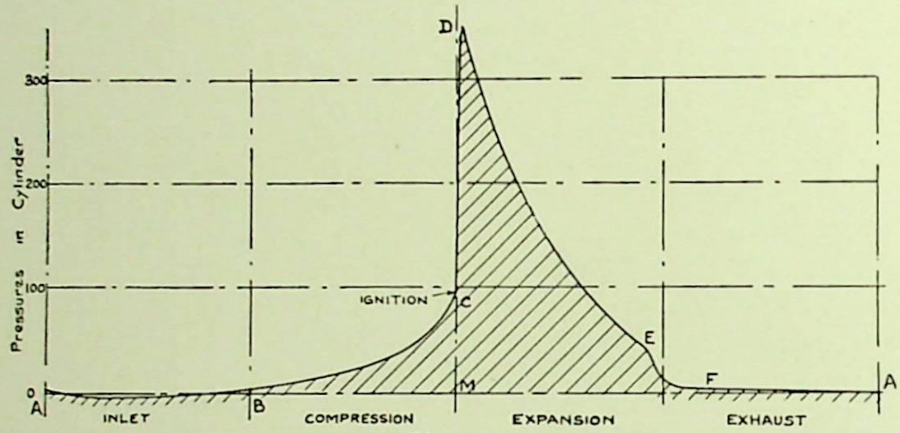


Fig. 14.

Denoting the four consecutive strokes of the cycle by the lines marked "Inlet," "Compression," "Expansion," and "Exhaust," in Fig. 14, and commencing as before with the inlet stroke, the pressure within the cylinder is a little below atmospheric, due to the suction effect of the piston as shown by the line AB. At the beginning of the compression stroke B, the pressure begins to increase in the cylinder, as the line BC denotes, until at C the pressure within the cylinder reaches the value shown by the vertical distance CM (usually from 75 to 90 lb. per sq. inch). At this point ignition occurs, and the pressure rises very rapidly to D, along CD, its value DE being about 250 to 350 lb. per sq. inch. At D, combustion of the petrol in the mixture is completed and the intense heat causes expansion of the gases along the line DE, the pressure dropping all the while until at the point E the exhaust valve opens, and along EFA the pressure drops to atmospheric value, which is represented by the level of the line AA

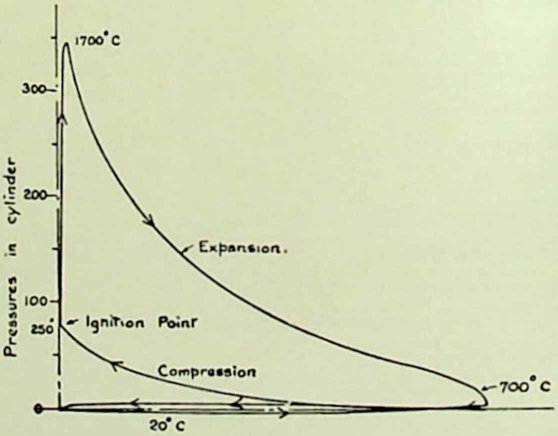


Fig. 15.

Another method, which is used by experimenters and also engine designers, of denoting the cylinder pressures, is to superimpose the lines AB, BM, ME, and EA upon one another, reversing BM and EA in the process so as to simulate the backward and forward strokes of the piston; in this manner a diagram of cylinder pressures, termed an *Indicator Diagram*, is obtained similar to that shown in Fig. 15, which corresponds with Fig. 14. Instruments known as *engine indicators* are now employed for automatically recording the pressures within the cylinder, in the form of indicator diagrams similar to the one shown.

Fig. 15 shows also the temperature values which exist in the cylinder-combustion chamber during the cycle of operations. It will be seen that the temperatures commence at the atmospheric value of 20° C., and owing to the heat of compression increase to 250° C. The highest temperature is attained at about the moment of highest pressure, and in this case is 1,700° C., falling progressively thereafter to 700° C. when the exhaust valve opens.

When it is remembered that 1,700° C. is the fusing temperature of platinum, and 1,500° that of steel, whilst pure aluminium melts at 660° C., the very high temperatures existing within the cylinder will be appreciated. In this connection it may be asked how it is that since the maximum temperature is higher than the melting point of steel, the cylinder, valves, and piston can withstand it. The answer is that this is an instantaneous temperature, and that in order to melt a metal this temperature would have to exist for a long period of time; moreover, the surfaces inside the cylinder are usually covered with a thin layer of carbon deposit, which is a bad heat conductor, and thus serves as a protection to the metal. Further, there is the cooling water or radiating fins which conduct the surplus heat away very effectively.

In the case of many early engines we have seen it stated that occasionally the pistons and cylinders were actually at a dull red heat; this can be appreciated after what has been mentioned in connection with the high temperatures existing, and bearing in mind the very inadequate cooling means usually provided in these early designs.

THE COMBUSTION CHAMBER. TURBULENCE

Having set forth the various stages of the working cycle of a typical engine working upon the four-stroke principle, and considered the question of the pressures and temperatures which exist within the cylinder, a short reference will be made to the somewhat complicated actions which occur within the combustion chamber.

When the fresh charge is drawn into the cylinder, during the suction stroke, in the case of an engine working at 2,000 r.p.m., the total time occupied by this stroke is only about $\frac{1}{70}$ sec., so that the mixture must travel through the inlet valve at a very high velocity in order to fill the cylinder in such a short interval. It is quite usual for the mixture to pass through the inlet pipe at a speed of 100 feet per second, that is, at about 70 miles per hour, during the suction stroke. These very high velocities, fortunately, produce considerable eddying, or turbulent effects, upon the mixture during its entry, and afterwards on the compression stroke; the mixture is thus in a state of turbulence when the ignition occurs. Owing to this turbulence, the combustion of the fuel particles is extremely rapid, for their motion spreads the flame at a very high speed or rate. If it were not for this turbulence, the high engine revolutions attained in modern engines would be impossible. It is also interesting to note that the combustion chambers of recent petrol engines have been specially designed in order to obtain increased turbulence.

The heat generated during the very brief explosion period (which is about $\frac{1}{100}$ to $\frac{1}{300}$ sec. in a modern high-speed engine) is used up or " absorbed " in several ways.

In the first place, owing to the cylinder walls, the piston valves, and combustion-chamber surfaces being at a comparatively low temperature, part of the heat of combustion is conducted through the metal of these parts, the greater portion passing to the water jacket, or in the case of an air-cooled engine to the air, *via* the cylinder-cooling fins.

Another portion of the heat is carried away by the hot exhaust gases and is lost in the outside atmosphere. We have already seen that the exhaust temperature is about 700° C. This is sufficient to cause a badly designed or partially choked engine silencer pipe to become red-hot.

The balance, or remainder of the heat of explosion, is the portion which is usefully employed in expanding the burnt gases and in doing useful work on the piston. In this connection it may be of interest to quote the following analysis of the results of some tests which were made upon a petrol engine. If the total amount of heat obtained within the cylinder by the combustion of the petrol and oxygen of the air in the mixture be denoted by 100, then this heat was found to be utilised as follows, namely :

Heat usefully employed as brake horse-power in the engine . .	24·0
Heat used up in overcoming the friction of the moving parts of the engine	4·0
Heat carried off by the cooling water	28·0
Heat carried off by the exhaust gases	40·0
Heat radiated directly from the engine	4·0
Total . . .	100·0

It will thus be apparent that for every gallon of petrol used by a petrol engine, only about 24 per cent., that is, about one quart, is usefully employed in giving horse-power for driving, the other three quarts being wasted.

This appears to be a rather regrettable state of affairs, but a careful consideration of the principles has shown conclusively that it is neither practical nor possible to get much more useful work out of the fuel with this type of engine.

Compared with the best steam engine, which utilises only about 15 per cent. of the energy of the fuel employed in the boiler, the petrol engine is considerably more efficient ; but it is inferior to the Diesel type of engine, to which we shall refer later, and in which about 30 to 36 per cent. of the fuel is usefully employed. Another even more efficient engine is the Still engine, which is a combination of an internal-combustion and a steam engine. In this engine, on one side of the piston is an *explosion* engine and on the other a steam engine, the steam being generated by part of the exhaust gas and cooling-water heat of the explosion engine; in this case from 35 to 45 per cent. of the fuel's heat is usefully given out as horse-power.

THE TWO-STROKE ENGINE

Reference has been made already to the engine of Dugald Clerk, in which the cycle of operations is completed in two consecutive strokes. This cycle possesses the theoretical advantage that twice the number of power strokes are obtainable for a given engine speed than in the case of the four-stroke engine, so that if the pressures of explosion were the same about twice the horse-power could be obtained than in the case of the four-stroke engine of similar dimensions.

Before proceeding to a discussion of the actual results obtained, a brief description of the working of a typical two-stroke engine will be given, whilst detailed descriptions of typical motor-vehicle engines will form part of the subject matter of a later chapter.

Referring to the diagrams shown in Fig. 16, which illustrate the Enfield motor-cycle engine. Diagram No. 3 shows the engine with the piston at the top of the compression stroke at the moment of ignition. The peculiar shape of piston should be noted in this diagram. Referring to the lower side of the piston, it will be seen that there are three distinct openings into the cylinder as follows, namely, B, the *exhaust outlet*, C, the *inlet port* to the crankcase from the carburettor, and D, the *transfer port* to the cylinder from the crankcase. It is for these reasons that this type of engine is termed a *three-port* type of two-stroke.

Briefly speaking, the principle of this two-stroke engine is as follows: on the upward stroke of the piston the mixture from the carburettor is drawn into the crankcase by the suction effect created by the piston; on the downstroke of the piston it is compressed in the crankcase, until, when the piston is nearly at the bottom of its downward stroke, the transfer port is uncovered and the mixture flows into the cylinder, there to be compressed on the next upward stroke, and fired. The exhaust gases escape through the port B, which is uncovered by the piston as it descends on the firing stroke.

In Fig. 16 (3) the mixture (as indicated by the arrows) is flowing into the crankcase under the influence of the suction of the piston moving upwards; at the same time the mixture above the piston is being compressed, and when the piston is at the top of its stroke, fired. In the next diagram (Fig. 16 (4)) the piston is descending on the firing stroke and the exhaust gases are shown escaping from the port B, just uncovered by the piston. It will be noted that the transfer port D from the crankcase (in which the mixture drawn in during the previous stroke is now compressed) is just about to be uncovered by the piston. In Fig. 16 (1) the port D is shown uncovered, and the path of the entering mixture can be traced by the arrows on the right-hand side ; some of the remaining

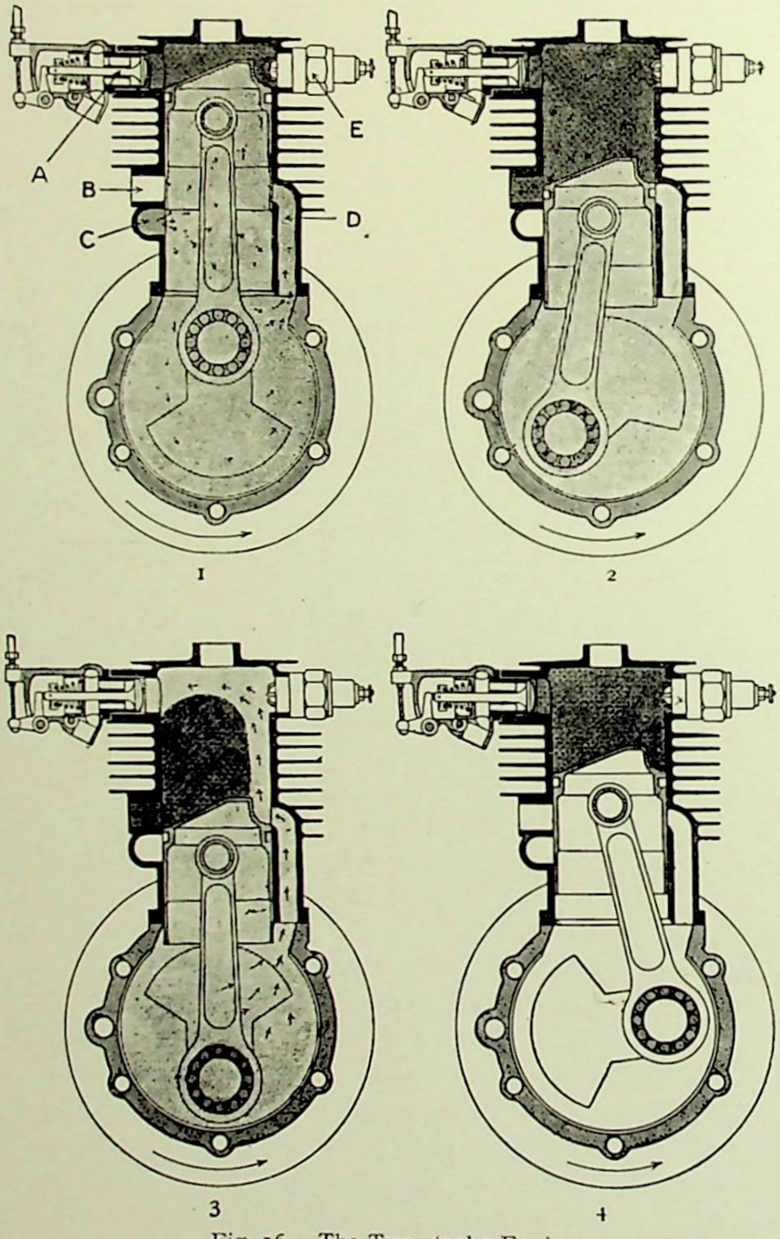


Fig. 16.—The Two-stroke Engine.

exhaust gases are still escaping through B. In Fig. 16 (2) the cylinder is closed and the ascending piston is shown compressing the mixture above it ; a partial vacuum is being formed at the same time in the crankcase, and as soon as the piston has ascended a certain distance and is nearing the top of its stroke, the inlet port C is uncovered, and the fresh mixture flows into the crankcase under this suction effect. The sparking plug is shown at E, whilst the valve at A is simply a means for releasing the compression in the cylinder and thus preventing the engine from working. The valve A opens inwards and allows the mixture to flow out into the atmosphere or silencer, instead of being compressed. It is a means of engine control employed on most motor-cycle two-stroke engines.

It will be apparent from what has been mentioned that this engine has no poppet valves, or valve mechanism, and that the piston performs the inlet and exhaust operations by uncovering ports in the cylinder wall. Such an engine is comparatively cheap to make, and requires very little attention or adjustment. It is essential to keep the crankcase compression-tight, and periodically to cleanse the ports in the cylinder walls from the carbon deposit formed.

The method of lubrication employed in most two-stroke engines consists in mixing one part of lubrication oil to about sixteen parts of petrol and utilising the mixture in the carburettor in a similar manner to petrol. This system (known as the *petroil* one) is quite satisfactory in practice, and no troubles are experienced if the carburettor is cleaned out at fairly long intervals, and the crankcase drained of the small amount of oil which collects there after long periods of usage.

THE TWO-CYCLE ENGINE'S PERFORMANCE

Fig. 17 illustrates a typical diagram of operations of the two-cycle engine ; the points at which the various operations occur are marked on the circle denoting the path of the crankpin. The arrow denotes the direction of the rotation, X being the top dead-centre and Y the bottom. When the piston has descended to the position shown by the crank at A, the exhaust port opens and remains open for a period AB of 120°. The inlet period DC is 95° in the case illustrated.

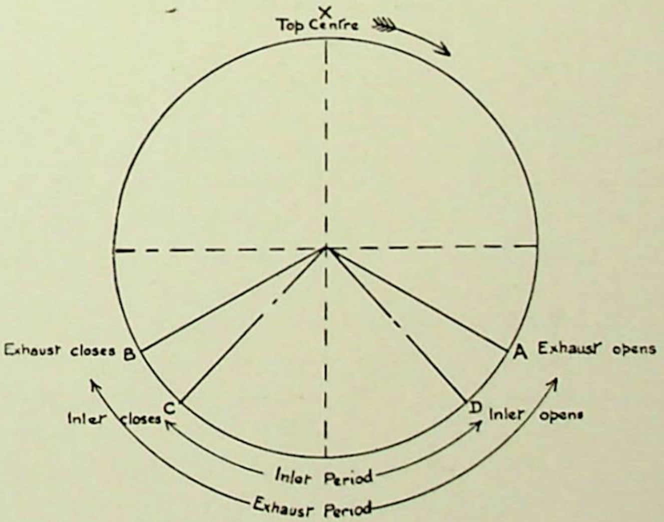


Fig. 17.

The pressures within the engine cylinder are usually appreciably lower than in the case of the four-stroke engine on account of the shorter period of mixture induction, and of the smaller quantity of mixture which is drawn in. Further, there is a partial escape of the mixture from the transfer port through the exhaust port, for both ports are open at the same time, as will be seen from Fig. 16 (1).

It is this charge-loss, amounting as it does in some cases to as much as 25 per cent. of the mixture, which is responsible for the greater fuel consumption of this type of engine.

Attempts have been made to improve the efficiency of the two-stroke engine by preventing this loss of mixture, and by utilising an air-blower or *scavenger* for sweeping out the burnt exhaust gases at the end of the firing stroke. Poppet and rotary valves have been employed to regulate the admission of the mixture into the cylinder, and in one or two cases the loss of charge has been prevented by the method of compressing air only in the crankcase, transferring this air to the cylinder, and compressing it in the usual manner, and by injecting the fuel into the cylinder head just before ignition, by means of a pump. Improvement has certainly occurred to a degree, due to these various attempts, but the fact still remains that the two-stroke engine is still inferior to the four-stroke engine from the point of view of the mean pressures developed.

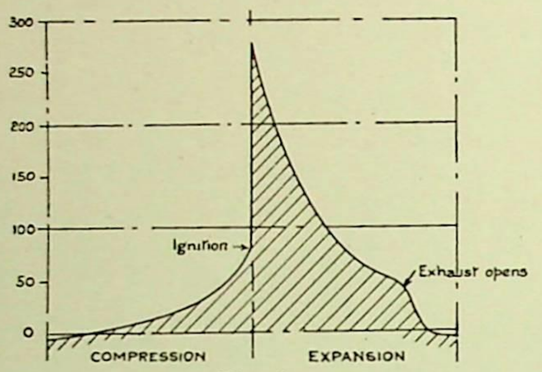


Fig. 18.

Theoretically, one should obtain double the power from a two-stroke engine of the same dimensions, and running at the same speed, as a four-stroke. In the case of slow-running, large, heavy-oil, and Diesel engines this is very nearly the case, for the fresh mixture or air has plenty of time to fill completely the cylinder. In the faster-running petrol engines, there is usually not sufficient time in which to fill the cylinder with mixture and to get rid of the burnt gases, which tend to dilute the mixture; consequently instead of getting twice the power, one only obtains from 50 to 70 per cent. more power.

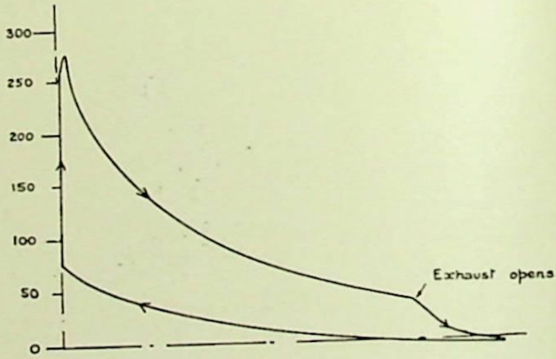
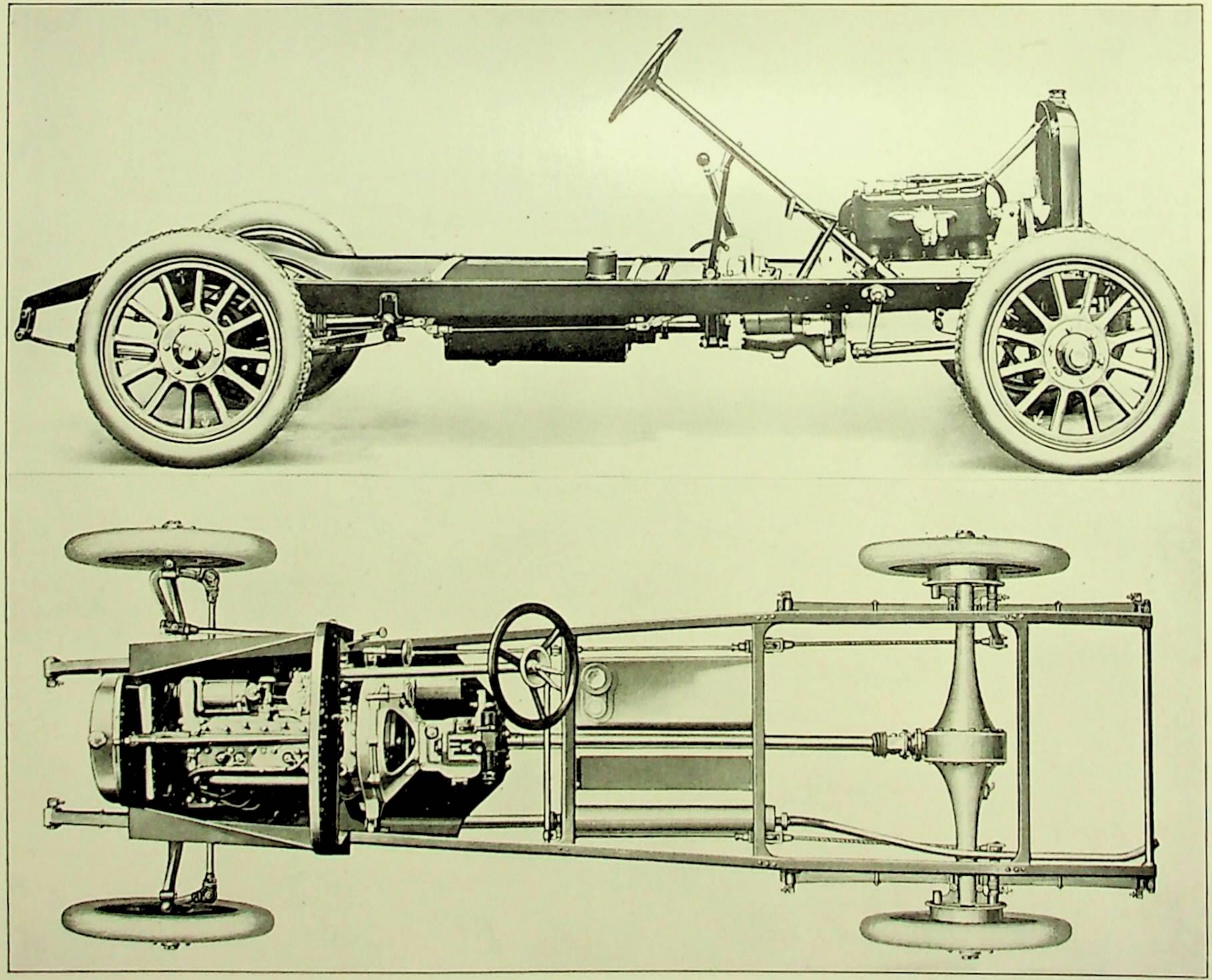


Fig. 19.

The diagrams of pressure shown in Figs. 18 and 19 are similar to those previously given for the four-stroke engine, but there are now only two curves instead of four. It is interesting to note the manner in which the pressure drops to atmospheric towards the end of the expansion stroke, and also the idle period afterwards when both transfer and exhaust ports are open together. It is this idle period which helps to lower the average pressure throughout the cycle of operations.



THE 12 H.P. AUSTIN CHASSIS IN SIDE ELEVATION AND IN PLAN.

CHAPTER II

COMPONENTS OF THE PETROL ENGINE

THE principles and the working of the petrol engine have now been considered briefly ; we shall next refer to the various mechanical parts and components of modern engines of the different types employed upon motor vehicles, confining our considerations for the time being to the poppet-valve types of engine. It is further proposed to deal with the engine from the same point of view as we have dealt with the theoretical side, namely, from the combustion chamber first (which is the initial region of the power development) and to proceed to the crankshaft through the intermediary of the piston and connecting rod. This is the same course as the power transmission follows.

THE ENGINE CYLINDER

The cylinder is usually made of good close-grained cast iron, carefully bored parallel, and finally ground to a high degree of smoothness. The combustion-chamber space above the extreme range of the piston's movement can take a variety of shapes and forms depending upon the valve arrangements and other considerations. Fig. 20 illustrates a typical air-cooled type of engine

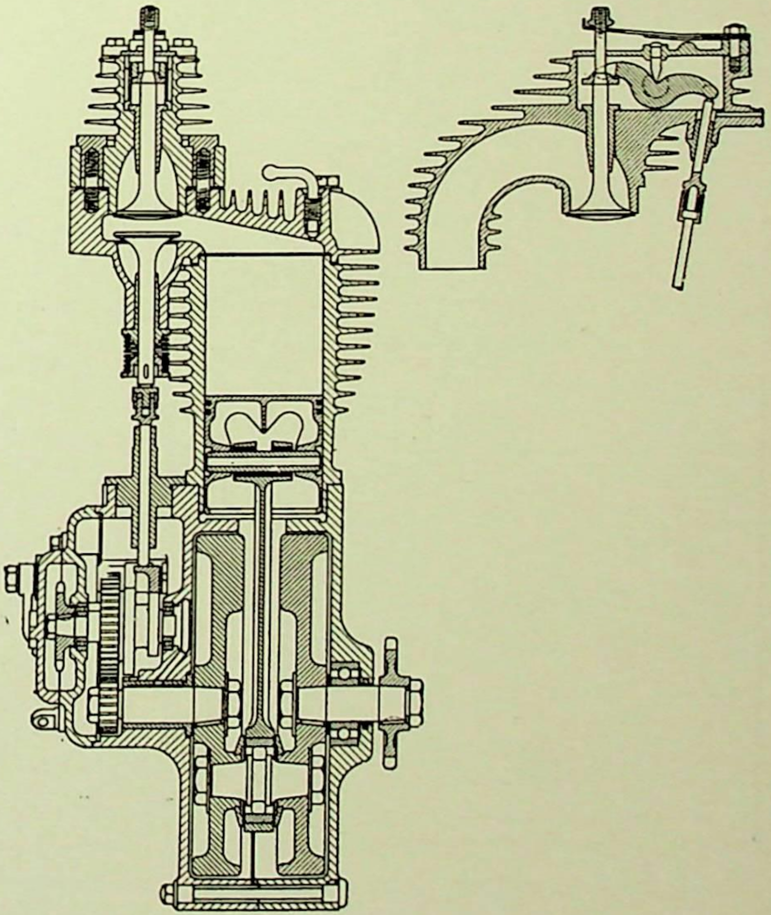


Fig. 20.—The Martinsyde Engine (*A.E.*).[1]

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[1] (*A.E.*) By courtesy of *The Automobile Engineer*.

cylinder made of cast iron, of the overhead exhaust-valve type, employed upon motor cycles. The exhaust valve is situated immediately above the inlet valve, and it will be noticed that the valves are situated in a pocket in the cylinder. The head of the cylinder, containing the exhaust valve and the compression tap, is separate from the barrel portion, this engine being of the type known as the *detachable head* one. The advantages of this type are that it enables the machining of the cylinder barrel to be more easily carried out, and that by removing a few holding-down bolts or nuts the whole top can be removed readily for cleaning out the carbon and getting at the valves.

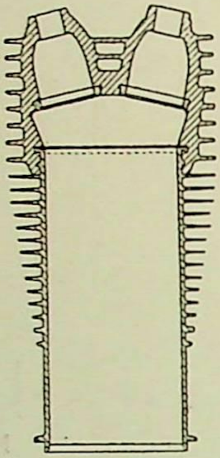


Fig. 21.

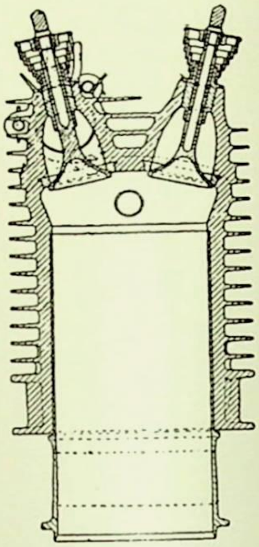


Fig. 22.

The cooling fins, it will be observed, are deepest at the combustion-chamber portion, where the greatest amount of heat is developed. Another type of air-cooled cylinder made of steel but fitted with an aluminium head is shown in Fig. 21. Although employed occasionally on high-speed motor-cycle and light-car engines, the proper domain of this type of cylinder is for aeroplane-engine work. These cylinders are machined all over, from a solid ingot of 40-ton steel, the thickness of the walls being only about $\frac{1}{10}$ to $\frac{1}{8}$ inch as compared with the $\frac{3}{16}$ to $\frac{1}{4}$ inch for cast-iron cylinders.

In some cases, cast aluminium-alloy cylinders are employed, in which the cylinder barrel is made of a steel liner shrunk into position and pinned in place. Fig. 22 illustrates a typical example. The better heat-conducting properties of aluminium enable a light and efficient cylinder construction to be obtained. A typical composition for a cylinder alloy is as follows: aluminium, 91 per cent.; copper, 7 per cent.; and tin, 2 per cent.

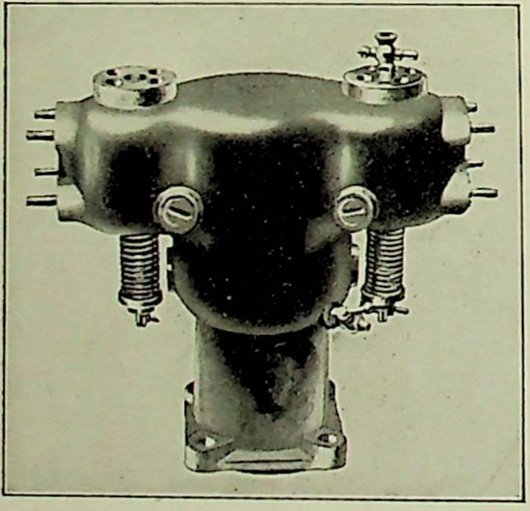


Fig. 23.—Example of a Tee-headed Cylinder.

The cylinder castings for water-cooled engines are, generally speaking, more complicated, due to the cooling-water passages having to be included in the casting. Fig. 23 illustrates a typical single-cylinder engine casting, the inlet and exhaust valves being on opposite sides of the cylinder; this is known as the *Tee-headed* type, whereas the one illustrated in Fig. 20 is the *L-headed* one.

In the case of engines possessing more than one cylinder, the cylinders are now generally cast all in one block (when it is termed a *monobloc*), or are cast in pairs in the case of a four-cylinder engine, threes in the case of a six-cylinder type, and usually in fours in the case of the eight-cylinder Vee, or vertical types. Twelve-cylinder engines frequently consist of two rows of six cylinders inclined at an angle to one another.

Typical examples of monobloc engines are shown in Figs. 25 and 26 for three and four cylinders respectively.

The example shown in Fig. 26 illustrates the use of the detachable cylinder head in the case of a four-cylinder engine. In this case the joint between the head and the barrel casting is made both gas- and water-tight by means of a gasket ring of thin copper with asbestos between the two faces of the copper. Needless to say, the shape of the gasket ring must be identical with that of the combustion head-cover faces.

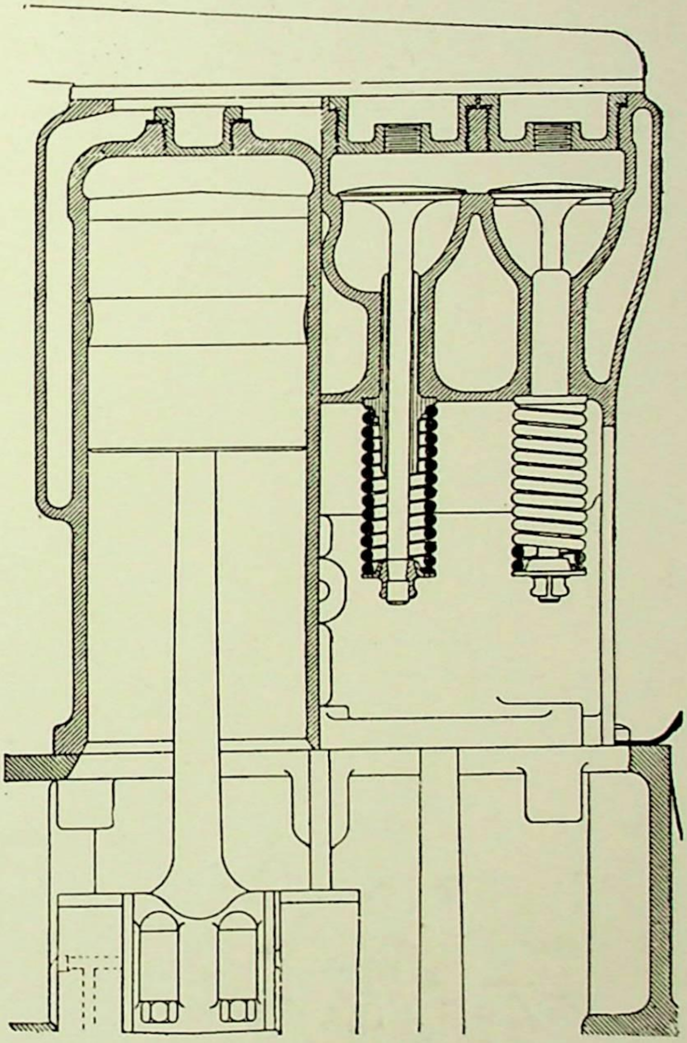


Fig. 24.—Inlet and Exhaust Valves Side by Side.

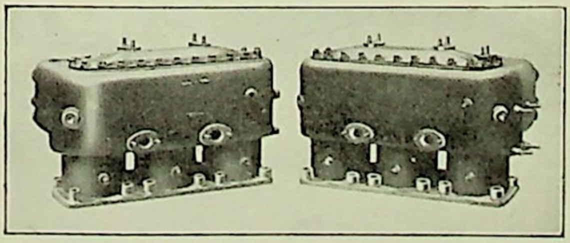


Fig. 25.—Monoblocs of Three Cylinders.

The monobloc type lends itself to mass production, and also results in a compact, clean cylinder block. On the other hand, for the owner who carries out his own repairs, the monobloc is cumbersome and

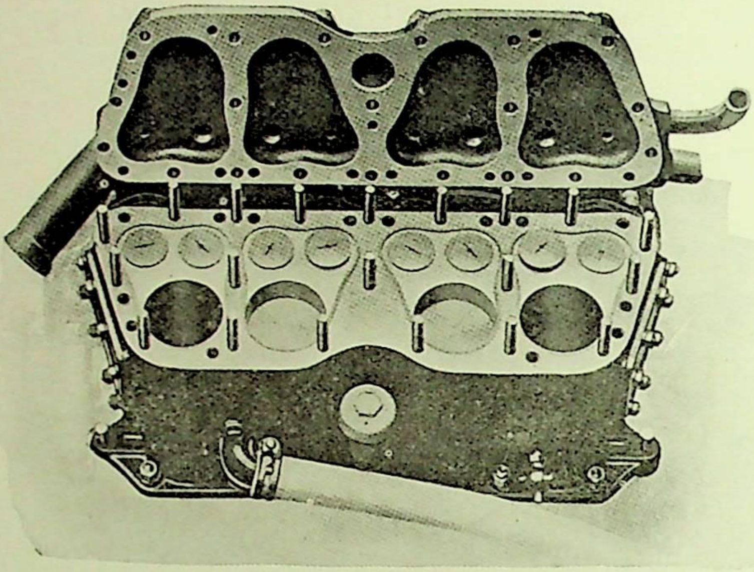


Fig. 26.—Showing Monobloc Cylinder with Detachable Head in the Packard Engine.

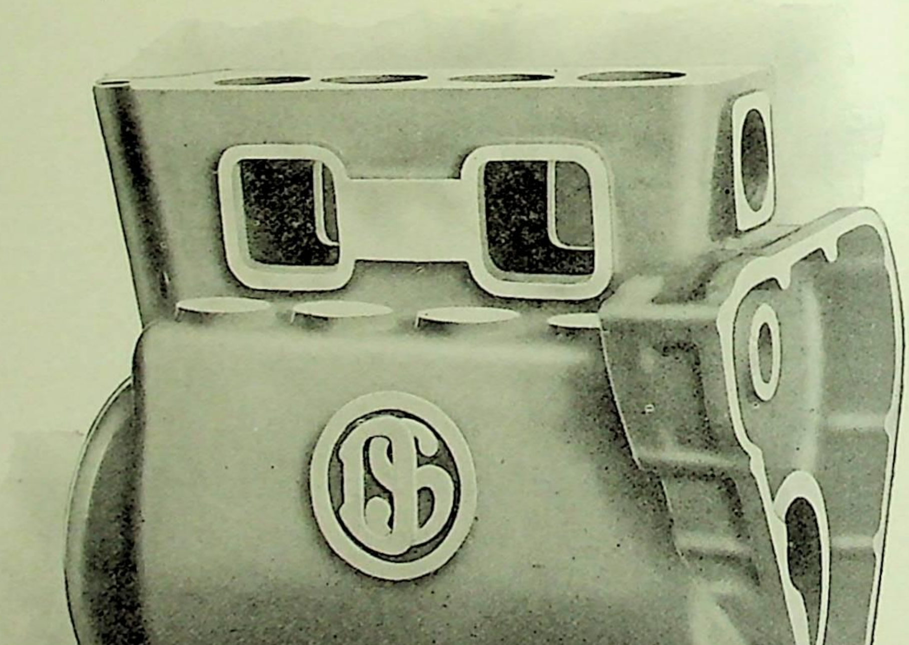


Fig. 27.—The Dorman 12 h.p. Engine Cylinder and Crankcase Casting.

heavy, making the removal and return of the cylinders a matter of some difficulty.

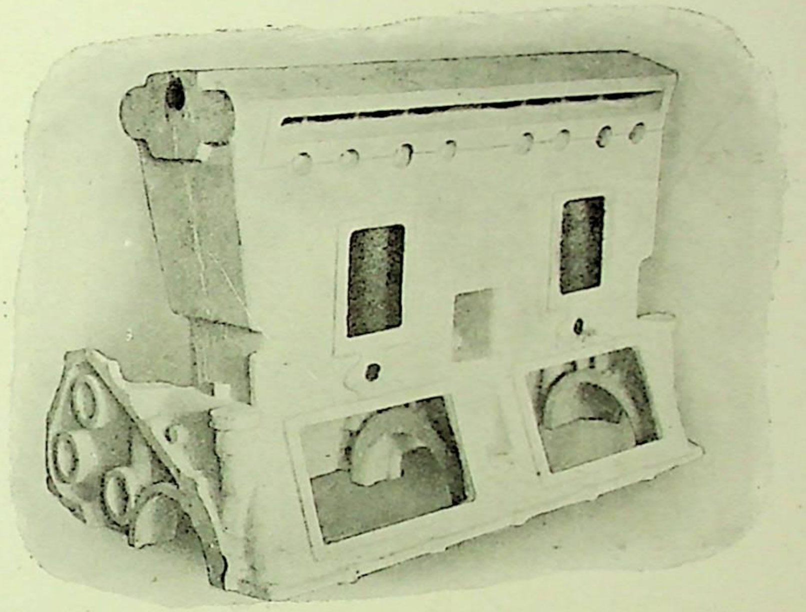


Fig. 28.

The modern tendency in engine design is to cast not only the cylinders but also part of the crankcase all in one, and thus to save the machining and fitting of the separate cylinders or cylinder blocks to the crankcase faces.

A typical casting, in an aluminium alloy, of this type is shown in the case of the four-cylinder Dorman 12 h.p. engine, in Fig. 27, and in which the cylinders, water jackets, crankcase, and timing casing are integral. Cast-iron liners, very accurately machined on both sides, are pressed into the jacket casing from the top, and a flange at the upper extremity of each beds down flush into a circumferential groove in the casting. Towards the lower end a shoulder is formed, and between the under-face of this and the bottom of the water jacket a rubber ring

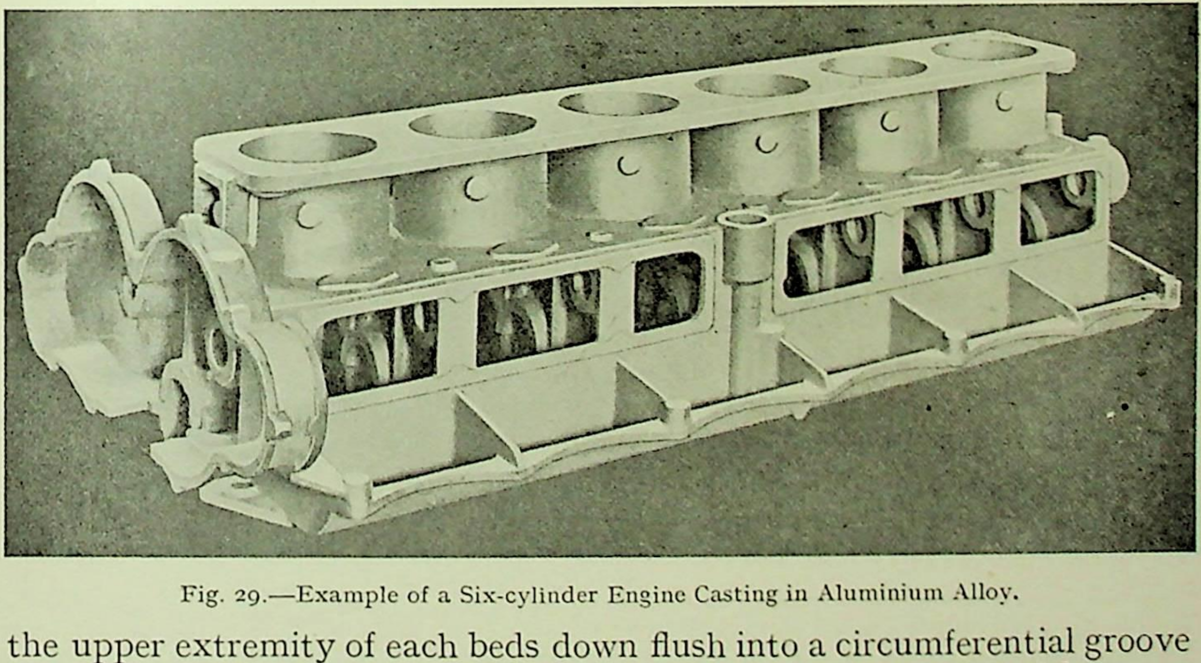


Fig. 29.—Example of a Six-cylinder Engine Casting in Aluminium Alloy.

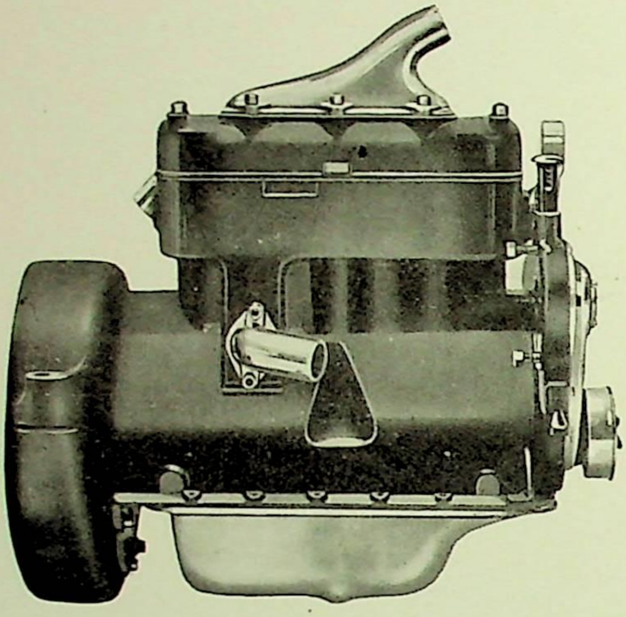


Fig. 30.

is held under compression. This ring not only makes a water-tight joint, but allows for the slight relative expansion of the two metals, aluminium and cast iron. A detachable head completes the cylinder at its upper extremity. Fig. 28 illustrates another example of a complete overhead valve type of cylinder casting.

A good example of a six-cylinder crankcase and cylinder barrel casting, by the Birmingham Casting Co., Ltd., is that shown in Fig. 29. The webs for holding the engine down to the frame or engine bearers, and also the bosses for the crankshaft and camshaft bearings, are clearly discernible in the illustration. An example of a complete unit built up of (*a*) a detachable head, (*b*) cylinder and crankcase casting, and (*c*) pressed steel crankcase cover, is that of the Continental four-cylinder 13·2 h.p. engine illustrated in Figs. 30 and 31. In the former figure the uppermost portion is the circulating water outlet from the cylinder jackets. In the end view shown in Fig. 31 it will be noticed that the timing-case cover is a steel pressing; the engine bearers can also be observed in this view.

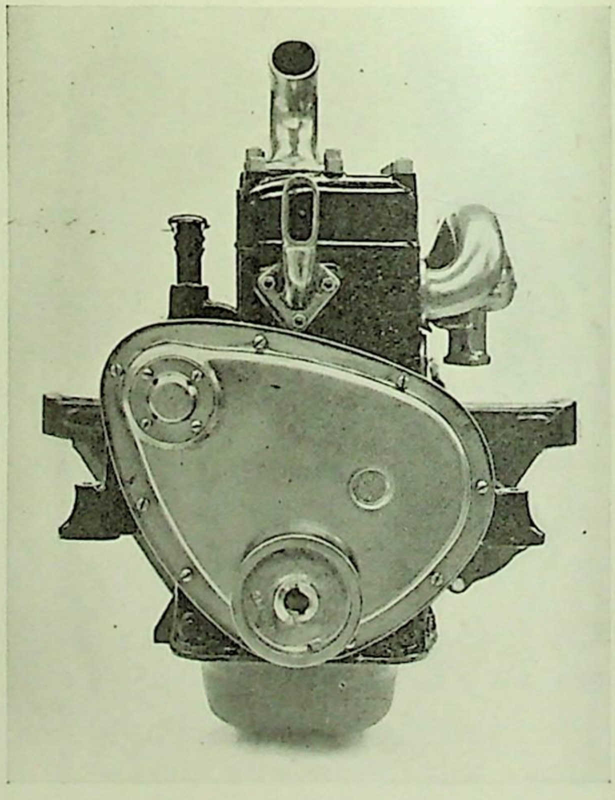


Fig. 31.

CYLINDER DESIGN

The shape of combustion head is governed by several theoretical and practical considerations, and the final result is generally a compromise between these. There is a variety of shapes of cylinder head in use at the present moment, the principal types being illustrated diagrammatically in

Fig. 32. From theoretical and experimental results the most efficient shape of combustion head is known to be the one giving the smallest amount of surface for a given volume, that is to say, a spherical shape as shown in A; the exploding gases lose the minimum amount of heat to the walls for this shape. Fig. 32, B, illustrates the practical interpretation of this type, both inlet and exhaust valves being in the head. The next approach to the ideal type, and one which is frequently adopted for facilitation of manufacturing processes, is that shown in Fig. 32, C. The L-headed form shown in E is widely employed and is considered superior to the tee-headed type shown in D, as greater turbulence and a shorter flame-travel path are obtained. The shape shown in F, in which the inlet is over the exhaust, has been employed on some car-engines and is probably rather better than E; it is very common on motor-cycle engines. There is now little doubt that the overhead valve types shown in B and C are more efficient from the power-output standpoint, as conclusive tests have shown.

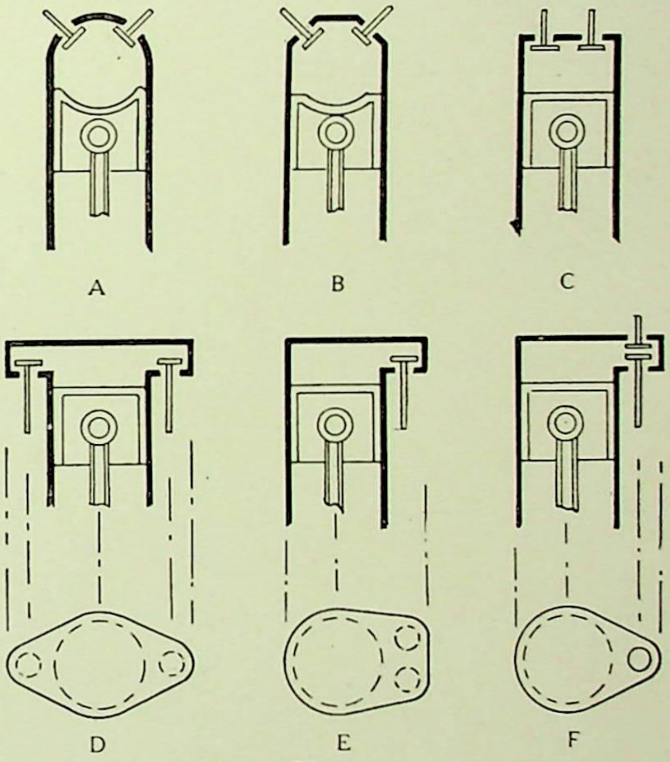


Fig. 32.

The dimensions of the cylinder are fixed by the required power of the engine. Thus the area of the cylinder bore determines the total force exerted on the piston, other conditions remaining unchanged.

If H.P. denotes the horse-power of the engine (at the piston)

A the cylinder area, or $\frac{\pi d^2}{4}$, where d is the bore in inches,

L the length of the piston's stroke in inches, and

N the number of revolutions of the crankshaft per minute,

then the horse-power h.p. $= \frac{p.\text{L.A.N.}}{792{,}000}$ for a four-stroke engine

$$= \frac{3 \cdot 1568}{10{,}000{,}000}\, \text{p.L.}d^2\text{N.}$$

Where p is the average pressure, in lb. per sq. inch in the cylinder. The value of p depends chiefly upon the compression pressure, and may be taken as being approximately 1·2 times the compression pressure.

From the formula given, the bore d of the cylinder can be estimated if a certain engine speed, say 1,500 r.p.m., be assumed.

The minimum thickness of the cylinder walls should not be less than that given by the relation:

$$t = \frac{pd}{2f} \text{ inches,}$$

where f is the safe stress for the material of the walls, which may be taken as 2,000 and 3,500 lb. per sq. inch, respectively, for cast iron and steel.

The minimum thickness should be greater than $\frac{3}{16}$ inch in the case of cast iron, for practical considerations connected with the casting and machining. An empirical formula in use in America is as follows:

$$t = \frac{d}{32} + \frac{1}{8} \text{ (inch).}$$

The water-jacket space width should not be less than $\frac{3}{8}$ inch for natural circulation and $\frac{5}{32}$ to $\frac{1}{4}$ inch for pump circulation; it is usual to divide the cylinder bore by the factor 6·6 in order to obtain the jacket width.

Petrol-engine cylinders are now invariably tested hydraulically before being fitted to new engines, and after they have been machined; the hydraulic pressure employed is usually greater than 500 lb. per sq. inch; and for water jackets 50 to 60 lb. per sq. inch. Fig. 33 illustrates the form of hydraulic cylinder tester employed by Messrs. Dorman Long for testing their petrol-engine cylinders to a pressure of 500 lb. per sq. inch, and the water jackets to 50 lb. per sq. inch. The pressure is applied by means of the hand-pump shown in the right-hand side, the pressure attained being indicated by the gauge.

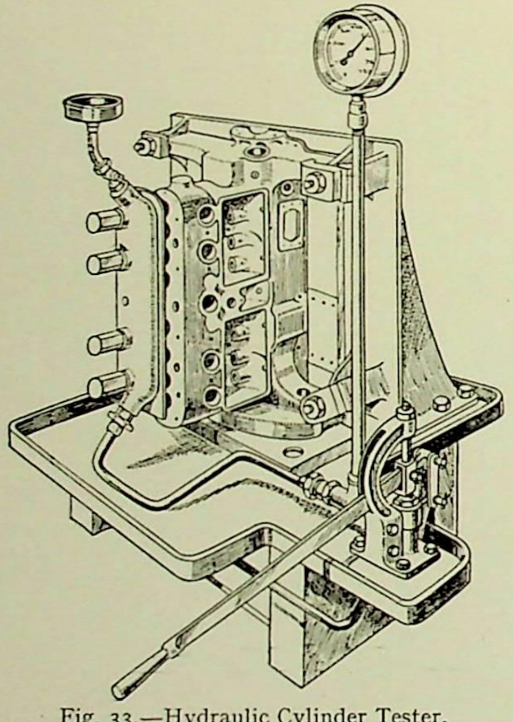


Fig. 33.—Hydraulic Cylinder Tester.

VALVES

Reference has been made already to the subject of poppet valves and to their use in the engine, and further, the different arrangements of the valves for high efficiency and for other reasons have been discussed.

Although, as we shall see later, there are other types of valves employed, the poppet valve is by far the most universal in modern engines, on account of its simplicity, cheapness, and reliability. A

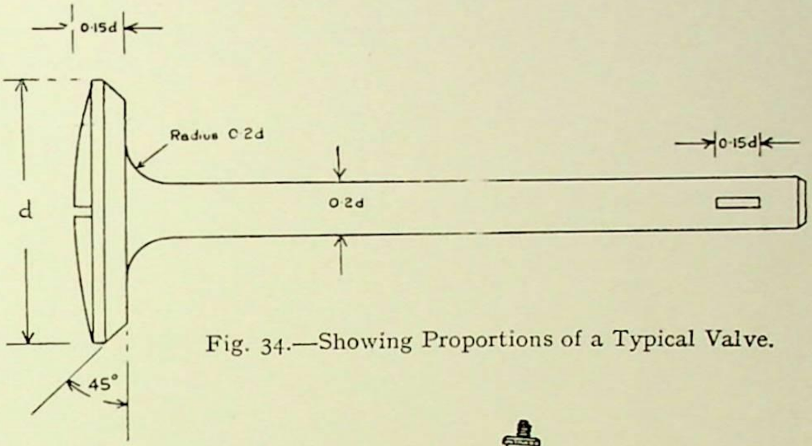


Fig. 34.—Showing Proportions of a Typical Valve.

typical valve, with approximately correct dimensions, is shown in Fig. 34. For this type of valve the lift is usually about one-quarter to one-fifth of the valve diameter at the lower portion, i.e. the port diameter.

The angle of the seating is generally made about 30° to 45°, although in some cases flat-seated valves are employed.

The size of the valve is determined by the piston diameter as a rule, but with the limitation that the gas velocity through the valve ports shall not exceed from 90 to 150 feet per second. The relation between the valve diameter and piston diameter may be taken as:

$$d = 0{\cdot}414 \,.\, \mathrm{D}$$

where d = diameter of valve and D = piston diameter in inches. Another relation derived from a consideration of equal flow through the valve and cylinder is as follows:

$$d^2 = \frac{\mathrm{D}^2 l \,.\, \mathrm{N}}{86{,}400} \text{ inches,}$$

where N = r.p.m., and l = stroke in inches, the gas velocity being taken at 120 feet per second.

Fig. 35 illustrates a complete valve system, showing the valve spring, collar, cotter, tappet, and cam, as employed on the Wolseley engines. Incidentally, this illustration shows an arrangement for reducing the valve noise, namely, a fibre pad placed between the tappet and the valve stem. The

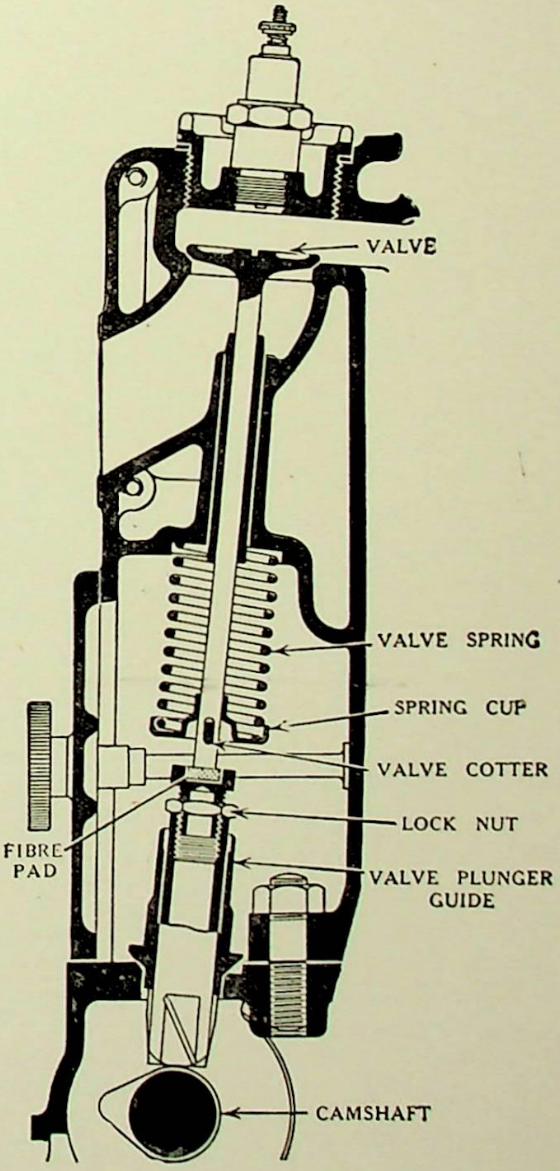


Fig. 35.—Typical Valve Components.

cover-plate on the left-hand side with the milled-headed screw is to enclose the valves for the purpose of keeping out dust, and for noise reduction.

VALVE TIMING

The opening and closing of the valves is an exceedingly important matter in petrol-engine practice ; it is essential that each valve should open at the correct moment, remain open for the correct period, and close again at exactly the precise moment. In the case of different designs of engine the manufacturers usually differ in their valve-timing systems, and adopt the best one for their own particular engine. The actual amounts of the valve openings are determined solely by the valve cam design, whilst the times of opening and closing are governed by the positions of these cams relatively to the engine crankshaft. Usually these times can be altered by displacing the driving wheels or chains, from the crankshaft to the camshaft, by one or more teeth.

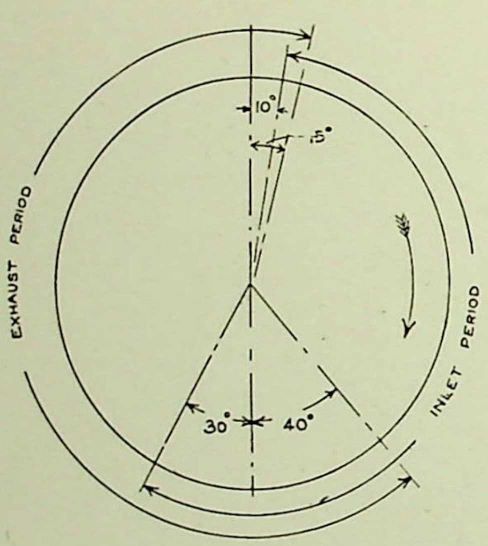


Fig. 36.

A typical valve-timing diagram (referred to the crankpin circle), for a modern high-speed car type of petrol engine is given in Fig. 36, and represents average practice amongst engine designers in this respect. For slow-speed engines the inlet valve should open at the top dead centre and close at the bottom dead centre, whilst the exhaust should open at 20° to 30° before bottom dead centre on the firing stroke and close at top dead centre.

In the case of engines designed to work at high speeds and to give great power output for a given size, the timing system is rather different, as the inertia of the moving gases influences the results. It is usual in such cases to open the inlet valve about 5° to 10° past top dead centre, and to close at 30° to 50° past bottom dead centre, in order to draw in as full a charge as possible. The exhaust valve should open at 30° to 50° before bottom dead centre on the firing stroke and close at 5° to 20° after top dead centre. This overlapping of exhaust and inlet often gives a cylinder scavenging effect, but generally speaking an engine with this valve timing is not so flexible as one with a timing arrangement similar to that shown in Fig. 36.

VALVE MATERIALS

Owing to the high temperatures to which the valves (more particularly the exhaust valve) are subjected, and to the rapid hammering under the action of the tappets and valve springs, special steels are employed which combine high resistance to temperature effect, and to corrosion (for the exhaust gases are corrosive), with high tensile strength.

The alloy steels are invariably employed for valves, tungsten, chromium (stainless), and nickel steels being the most common ones in use.

The following are typical compositions and tensile strengths of these steels, as recommended by the British Engineering Standards Committee :

1. TUNGSTEN STEEL

Material	Iron	Carbon	Manganese	Tungsten	Chromium
Percentage	81·4 to 81·55	·55 to ·70	·40 (max.)	14·0 (min.)	3·5 (min.)

The tensile strength of this steel should be about 50 tons per sq. inch, and the Brinell hardness 277, after being heat-treated.

2. HIGH CHROMIUM STEEL

Material	Iron	Carbon	Manganese	Chromium	Sulphur	Phosphorus
Percentage	87 to 93	·4 to ·7	·3 to ·6	8·0 to 12·0	·07 (max.)	·06 (max.)

The tensile strength should be about 50 to 75 tons per sq. inch, and the Brinell hardness between 192 and 269, after heat treatment.

3. NICKEL STEEL

Material	Iron	Carbon	Silicon.	Manganese.	Nickel.	Sulphur.	Phosphorus.
Percentage	95 to 97	·25 to ·35	·30 (max.)	·4 to ·7	2·5 to 3·5	·06 (max.)	·06 (max.)

The tensile strength should be about 45 to 55 tons per sq. inch, and the Brinell hardness 220, after heat treatment.

VALVE OPERATION

The two principal valve systems commonly employed are the direct and inverted (or overhead) ones. In the former case the valves are directly operated from the camshaft, as shown in Fig. 35 ; in the latter case the valves are operated either by push rods and overhead rockers, similar to those shown in Fig. 10, or are operated from a camshaft situated along the tops of the cylinders and driven by bevel

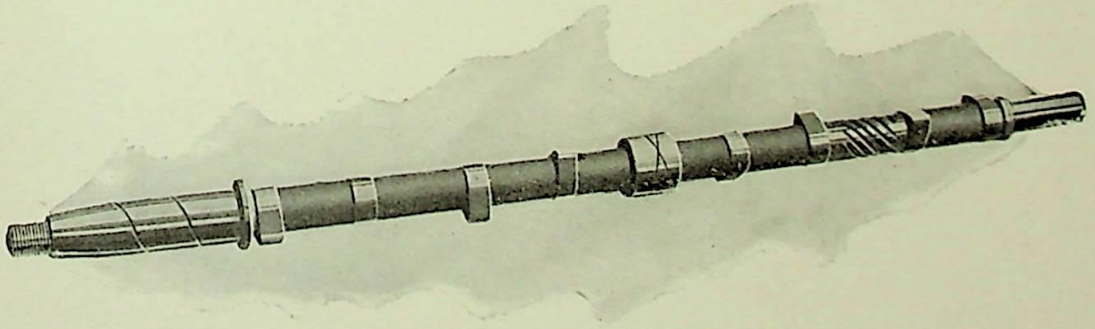


Fig. 37.—A Camshaft of the Solid Type.

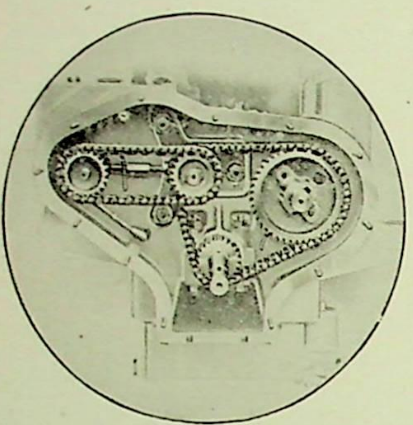


Fig. 38.

or spiral gearing from the engine crankshaft. In the direct type the valves in multi-cylinder engines are usually operated from a single camshaft, parallel with the crankshaft, the cams being cut integral with the shaft and the whole machined from the solid as shown in Fig. 37. In the best practice, the camshaft runs in three ball-bearings, in the case of a four-cylinder engine, and is made of a case-hardened nickel, medium-carbon, or nickel-chromium steel.

The timing shaft is driven off the crankshaft by a two-to-one reduction gear of the

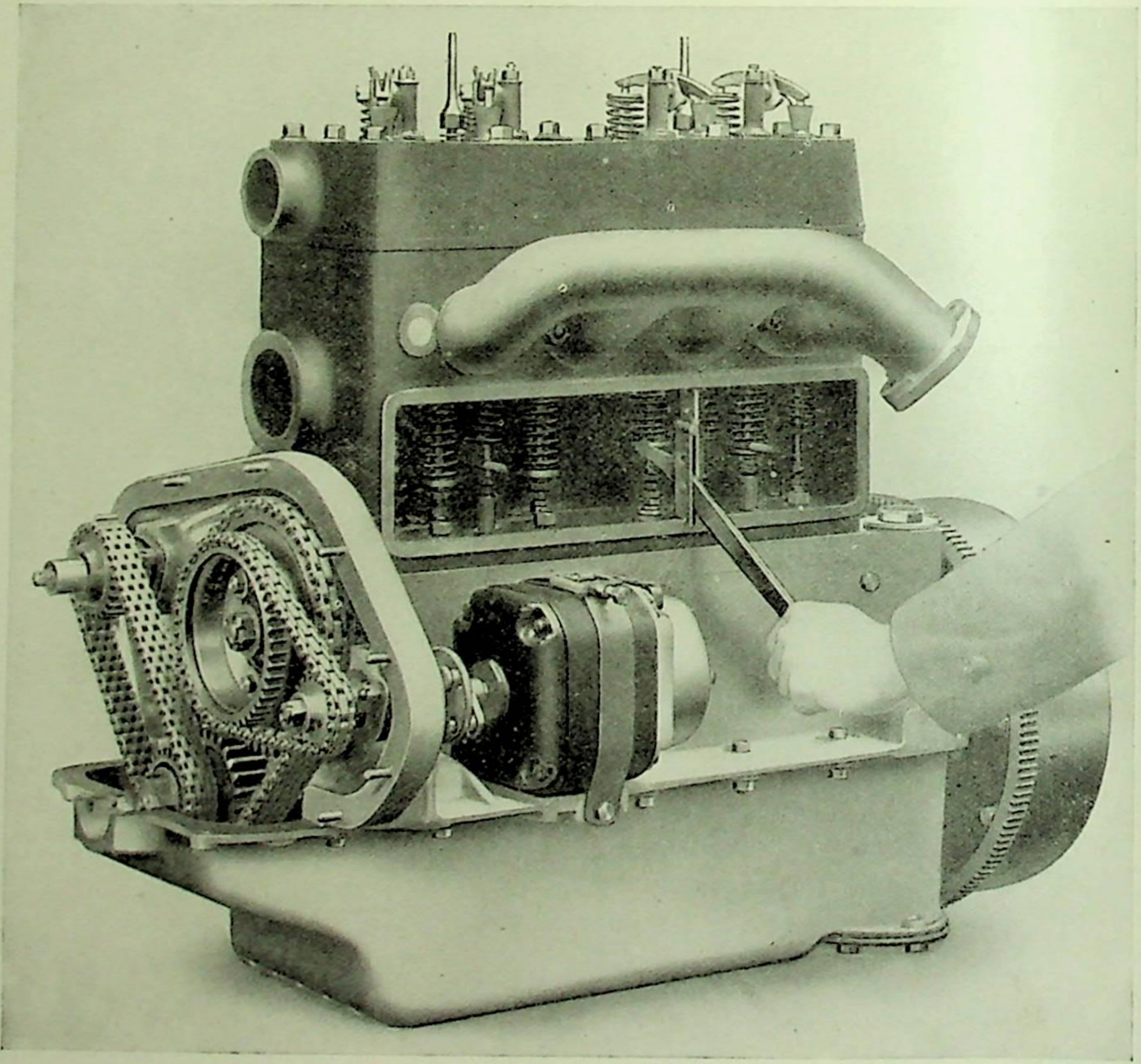


Fig. 39.—Chain-drive Examples on the Humber 15·9 Engine.

spur or chain type. The silent type of chain is widely employed for the latter purpose, and frequently drives not only the camshaft but also the magneto, water pump, and other accessories. Some typical examples of camshaft drives are shown in Figs. 38 and 39, whilst alternative methods of taking up slackness, and for adjustment, are given in Figs. 40 and 41.

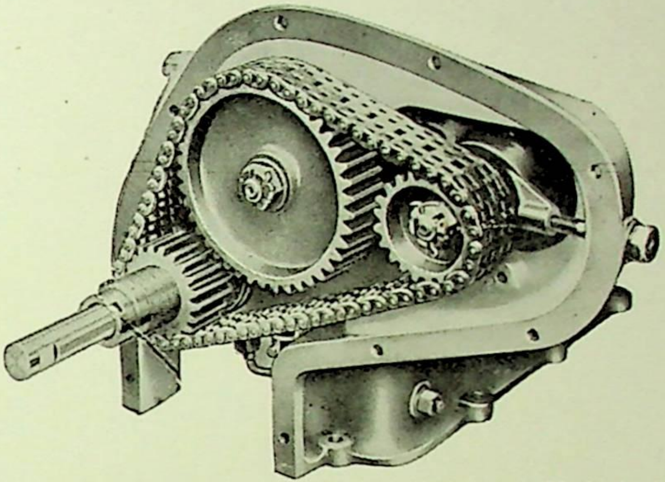
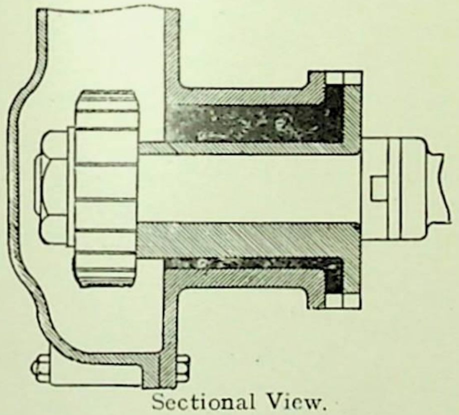


Fig. 40.—Showing Engine Timing Chain, and Method of Adjustment.

Fig. 39 illustrates the valve-operating mechanism of the 15·9 Humber engine. In this example the inlet valves are direct operated, and the exhaust valves of the overhead type operated by means of push rods and rocker arms. The method of compressing the valve spring for removal purposes is clearly shown in this illustration. It will also be observed that there are three separate silent chain drives, one from the engine to the camshaft with the usual two-to-one reduction, and a further drive from the camshaft to the lighting dynamo, shown attached to the crankcase; the left-hand chain drive, with no reduction, is for the magneto.


Sectional View.

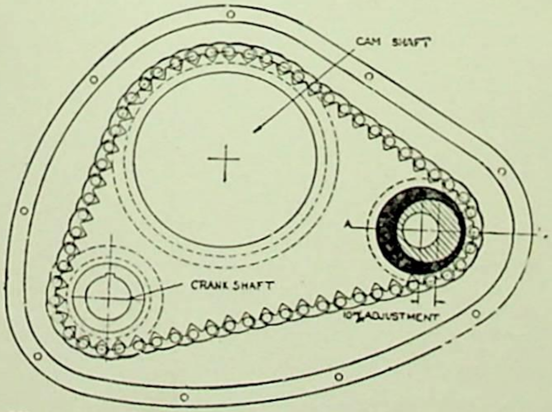


Fig. 41.—Showing Eccentric Type of Chain Adjusted.

An example of an overhead camshaft drive, by spiral bevel gearing, in the case of the Bentley engine, is shown in Fig. 42. Here a vertical shaft driven by spiral bevel gears engages with the bevel gear shown, which is rigidly attached to the overhead camshaft. Another example of an overhead camshaft is shown in the sectional illustrations in Fig. 43, which shows also part of the detachable cylinder head and the overhead valves and rockers. The engine depicted is of the four-cylinder type, and the camshaft is housed in the three bearings shown immediately under the milled screws holding the cover box in position. The rocker-arm shaft is shown immediately above the camshaft, with the rocker bearings in black.

An overhead camshaft for a six-cylinder engine is shown in Fig. 44, the drive being by worm and worm-wheel (the latter is shown on the extreme left of the diagram).

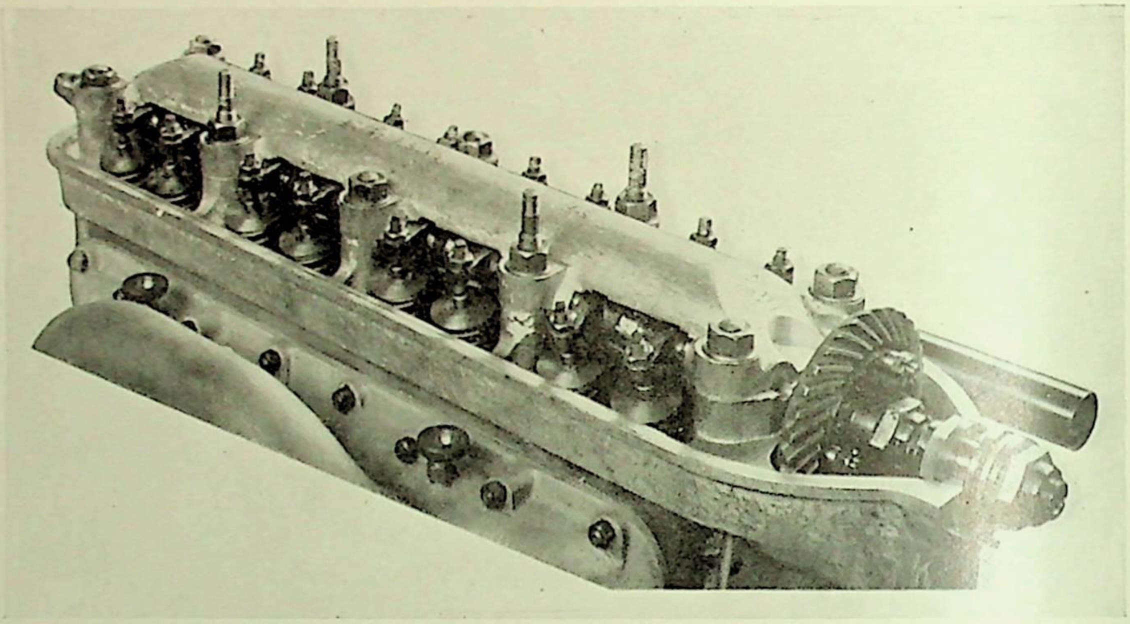


Fig. 42.—The Bentley Overhead Valve Gear.

A good example of overhead-valve design, on the push-rod and rocker-arm principle, is that of the Armstrong-Siddeley six-cylinder engine shown in Figs. 45 and 46.

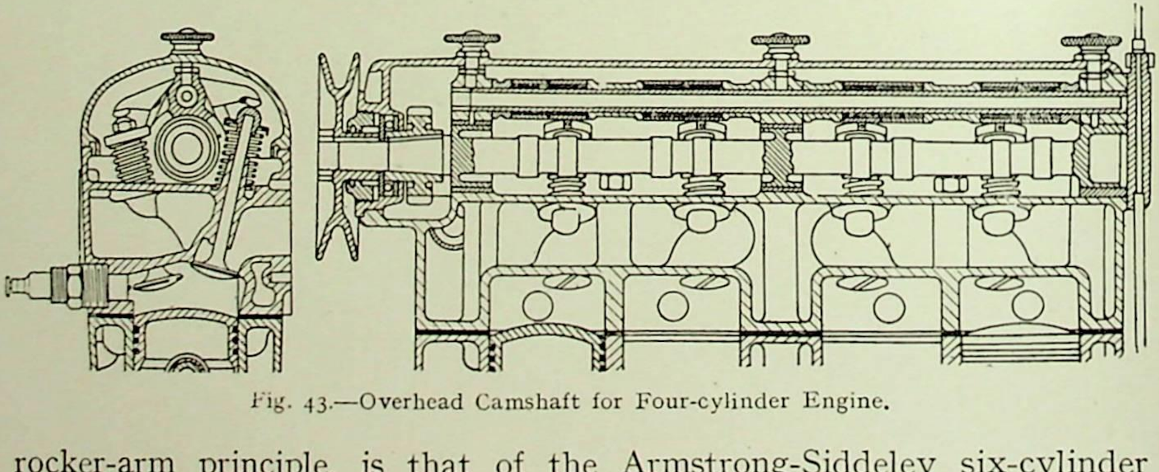


Fig. 43.—Overhead Camshaft for Four-cylinder Engine.

Fig. 46 depicts the cylinders (in monoblocs of three) with the valve casings removed so as to expose the valve heads and rocker arms. It will be seen that the ends of the latter are provided with balls which work in cups (on the one side), formed in the ends of the push rods, the other ends bearing directly upon the valve stems. Fig. 45 shows also the method of adjusting the clearance between the rocker arm and the valve stem, for it is desirable to keep this clearance as small as possible. In this case adjustment is made by means of the nuts on the push rod, locking nuts being provided to lock the adjusting nuts. The feeler gauge shown, held in the hand in Fig. 45, is for the purpose of adjusting the valve-stem clearances to the desired amount.

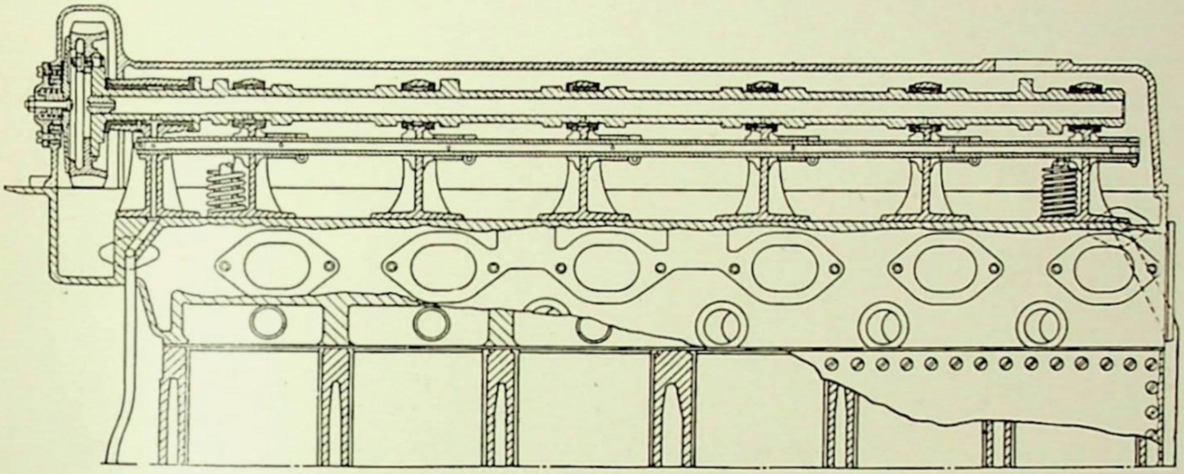


Fig. 44.—Overhead Camshaft on Six-cylinder Engine.

For overhead valves the clearance when the engine is warm should be about $\frac{5}{1000}$ inch for the inlet and $\frac{7}{1000}$ inch for the exhaust. For direct-type valves these clearances are usually $\frac{3}{1000}$ and $\frac{4}{1000}$ inch respectively.

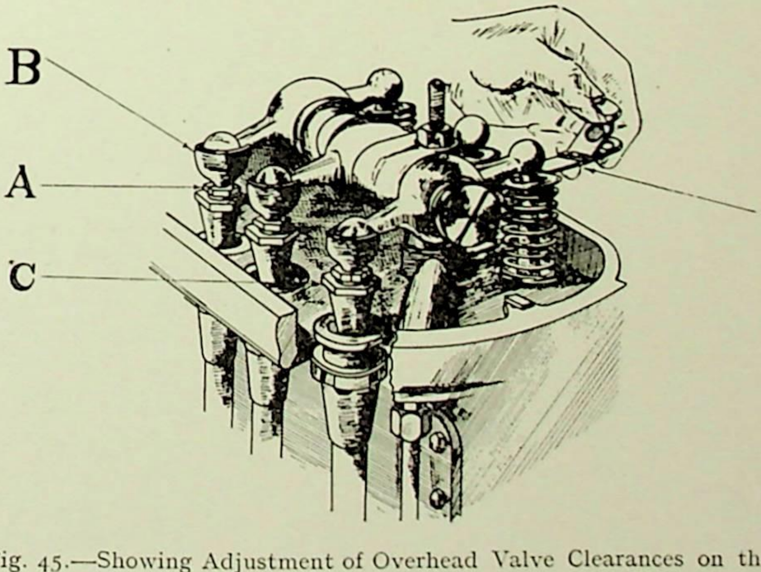


Fig. 45.—Showing Adjustment of Overhead Valve Clearances on the Armstrong-Siddeley Engine.

In connection with the push rods for overhead valves, it is commonly found that if the valve clearances are adjusted correctly when the engine is cold, they will be too great when the engine has warmed up, due to the relatively greater expansion of the cylinder head (which carries the rocker bearing upwards) than the cooler tappets. If, on the other hand, the clearances are adjusted when the engine is hot, then when cold there is often no clearance left, and the valves may even be slightly depressed from their seatings. To obviate this, it is usual to encase the push rods in tunnels cast in the cylinder, so that they will heat up and expand with the cylinder. An alternative automatic compensating system, known as the Jones, is shown in Fig. 47. There the rocker-arm bearing is eccentric to the rocker pin, and the small pin attached to this eccentric and to a tie-rod fixed at its lower end to the crankcase rotates the eccentric relatively to the rocker pin in such a manner as to keep the valve-stem clearance constant, always, as the cylinder expands or cools.

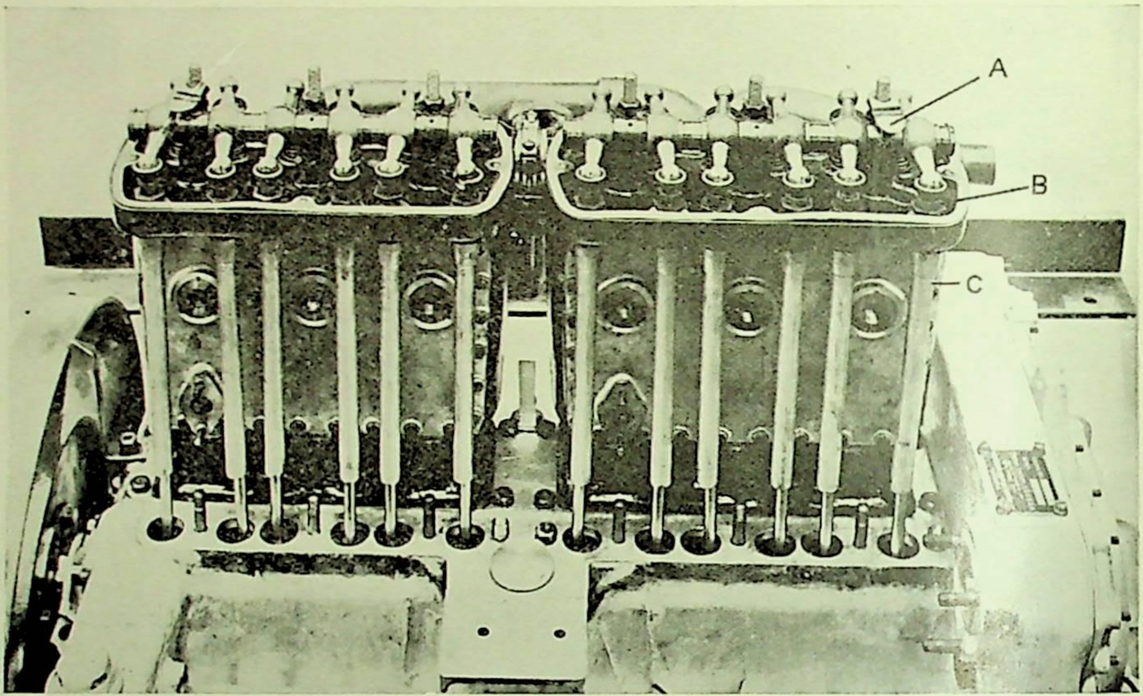


Fig. 46.—The Armstrong-Siddeley Overhead Valve Gear.

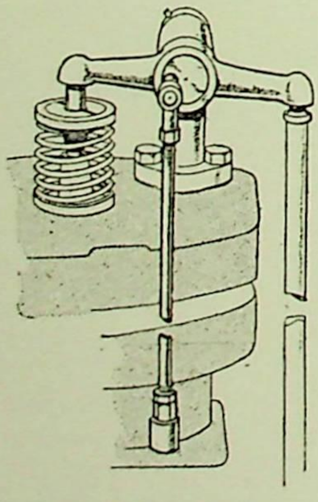


FIG. 47.—Automatic Compensating Device for Overhead Valve Tappet Rods (*Autocar*).

CAM SHAPES

The shape of the contour of the valve cam has an important influence upon the efficiency and running of the engine. The cam should be so designed that the valve is opened and closed as quickly as possible and remains open as long as can be allowed. There are two principal types of valve cam commonly employed, namely, the quick-lift and the gradual-lift types.

The former, which is illustrated in Fig. 48 (*a*), gives a fairly sharp opening and closing, with a comparatively long period of opening as shown by the length of the parallel portion of the valve-opening curve in Fig. 49 (*a*). The objection to this type of cam (which is employed chiefly in high-speed and racing engines) is the side-force effect on the valve tappet, due to the steepness of the cam, and the noise of the valve itself in suddenly opening and closing upon its seating.

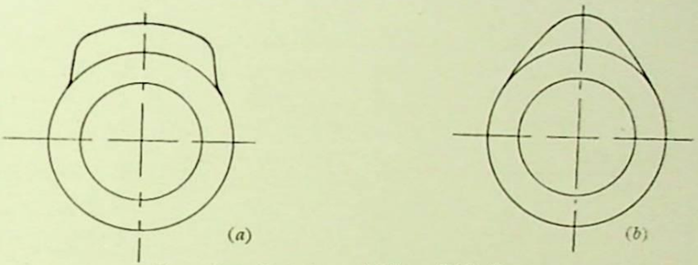
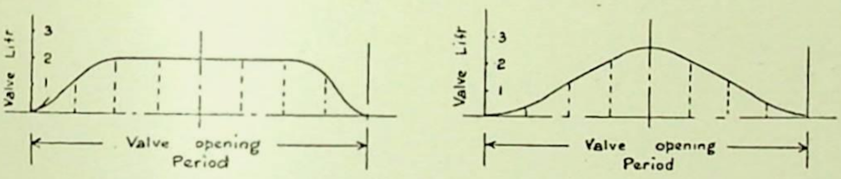


Fig. 48.—Quick and Slow Lift Cams.

The compromise generally adopted is to give the cam the shape shown in Fig. 49 (*b*); here the lift is more gradual, and the closing period also. The valve-opening curve is shown in Fig. 49 (*b*). The effective period is less than in the former case, but the valve is quieter and the valve mechanism wears much longer.



(*a*) Fig. 49. (*b*)

It should be pointed out that the amount of lift of the valve is determined by the maximum width of the cam, that is, by the greatest distance from the base circle in Fig. 48.

PISTONS

The function of the piston is to provide a gas-tight sliding member for transmitting the explosion and expansion pressure loads to the connecting rod and crank. The main desirable characteristics of the piston are that it should be a good sliding fit, well lubricated, as light as possible, and durable.

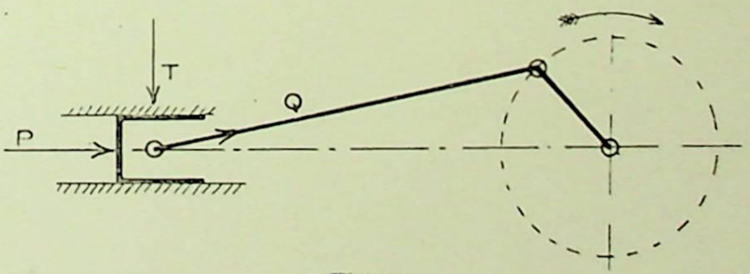


Fig. 50.

Fig. 50 is a line diagram showing the forces acting, in the case of a petrol engine, due to the pressure P upon the piston. It will be observed that there is a thrust T acting between the piston and cylinder walls, and a longitudinal force Q which creates a bearing pressure on the gudgeon pin. The piston surface must be adequate for taking this thrust, and experience has shown that if the length of the piston is equal to, or a little greater than, the piston diameter, the area will be sufficient. For slow-speed petrol engines requiring long service the length can be from 30 to 60 per cent. greater than the diameter. The bearing pressure on the piston should not exceed 25 lb. per sq. inch. The components of the piston are the main body, the rings, and the gudgeon pin with its locking device; the usual rotation employed in connection with pistons is indicated in Fig. 51.

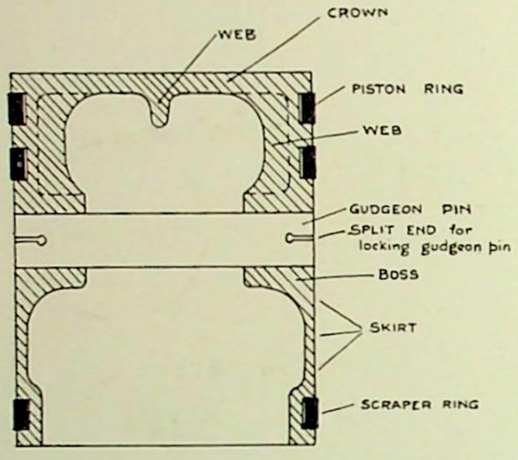


Fig. 51.

The majority of the pistons in use are made of a high grade of cast iron, since the wearing properties of this material are excellent; the piston rings themselves are also of cast iron. With cast-iron pistons it is usual to allow a clearance between the piston and the cylinder of $\frac{2}{1000}$ inch per one-inch diameter of the cylinder; thus the clearance for a $3\frac{1}{2}$-inch cylinder would be $\frac{7}{1000}$ inch, the piston diameter being 3·493 inches. The pistons are usually tapered slightly, the greatest clearance being at the top, which is the hotter end, and the difference is usually of the order of from two- to four-thousandths of an inch. There are three principal types of piston in general use, namely, the cast-iron type, the welded steel, and the aluminium-alloy types; the latter two are employed solely on account of their lightness, for, as will be shown later, it is desirable to reduce the weight of the piston and connecting rod to the smallest possible limits in order to obtain maximum engine speeds and power.

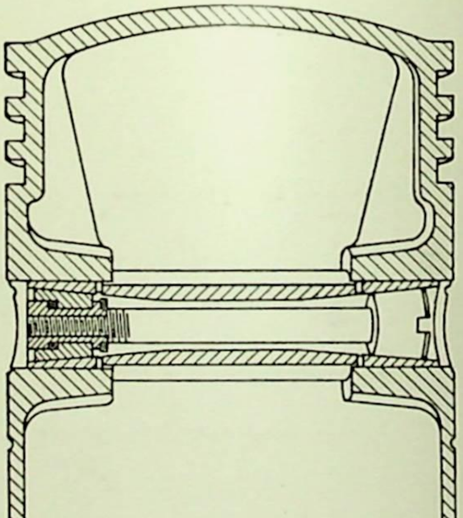


Fig. 52.—A Cast-iron Commercial Engine Piston.

THE CAST-IRON PISTON

This type is illustrated in Figs. 52 and 53. Fig. 52 is a good example of a piston designed for a motor-lorry engine, and it has a length approximately 60 per cent. greater than its diameter, in order to ensure good wearing service. It is provided with three cast-iron rings, the grooves for which are shown. The gudgeon pin is held in cylindrical bosses, cast with the piston, and braced rigidly in position by means of the webs shown. This gudgeon pin consists of two portions, namely, an outer sleeve which can rotate on an inner sleeve, which is locked in position by means of the conical-ended bolt and nut shown. The outer sleeve is a sliding fit in the bush of the connecting rod. Fig. 53 illustrates the Laystall type of light cast-iron piston, provided with a series of stiffening ribs from the crown to the walls, and gudgeon-pin bosses. Owing to the design of the ribs, this type of piston can be made appreciably lighter than the ordinary solid cast-iron type.

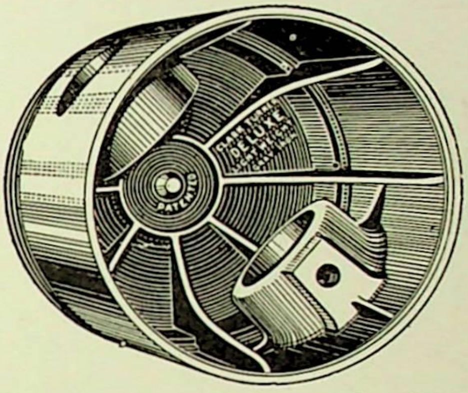


Fig. 53.—The Laystall Light Cast-iron Piston.

In the case of engines which have a tendency to oil the cylinder too freely, a piston ring is fitted below the gudgeon pin, and usually at the lower end of the skirt, so as to act as an oil-scraper ring. Alternatively, holes are drilled in the skirt itself, or in shallow grooves turned in the piston skirt.

The thickness of the crown of a cast-iron piston of the ordinary type is usually taken as that given by the relation—

$$t = \frac{1}{32}(D + 3) \text{ inches,}$$

where t is the thickness, and D the piston diameter. The smallest thickness should not, however, be less than $\frac{5}{32}$ inch.

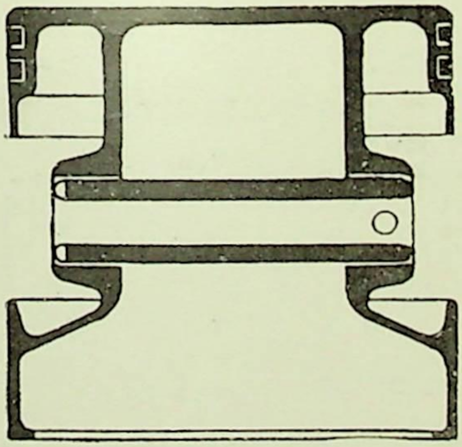


Fig. 54.—The Zephyr Welded-steel Piston, in Section.

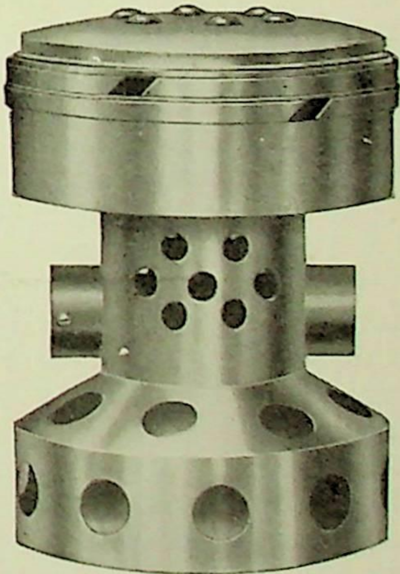


Fig. 55.—External View of the Zephyr Welded-steel Piston.

STEEL PISTONS

The difference between the tensile strengths of cast iron and steel is considerable, that of the former being about 7 tons per sq. inch, whilst good mild steel as used for pistons is about 35 to 40 tons per sq. inch. For this reason much lighter pistons, of the same or even greater strength than cast iron, may be made of steel, but as it is a somewhat difficult matter to obtain the correct shape, owing to the gudgeon-pin bosses and stiffeners required, it is usual to build up steel pistons by welding, screwing, or riveting.

A typical example of a satisfactory steel piston is the Zephyr type illustrated in Figs. 54 and 55. It will be seen that in this design the narrow waist of the piston carries the gudgeon pin, and the thrust from the head is taken direct, instead of passing through the walls. The radiation of heat is assisted by this design, resulting in the temperature of the top of the piston being reduced. Two rings only are fitted in the case of the Zephyr pistons used in the Dorman engines, as the length of the pistons is appreciably greater than their diameter, and the possibility of piston "slap" is avoided. In the case of racing types of petrol engine, the skirt is usually drilled out, to reduce the weight a little more.

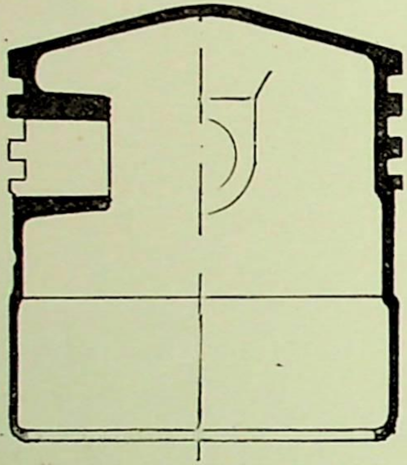


Fig. 56.—Pressed-steel Piston.

Pressed-steel pistons are sometimes used in petrol engines; and in these cases a saving of weight and increased strength are obtained over cast-iron pistons of the orthodox type. These pressed-steel pistons, which are made out of solid steel by powerful presses and dies, range in diameter from 2 to 10 inches, and are of a wide variety of designs. Examples of pressed-steel pistons are shown in Figs. 56 and 57, but, as we have already stated, they are but a few examples out of many.

It is interesting to describe the process of making a pressed-steel piston, for certain difficulties have to be overcome, namely, in connection with the gudgeon-pin bosses.

The pressing process resembles drop-forge pressing, due to the presence of the large bosses for the gudgeon pin. In order to permit of the withdrawal of the die from the interior of the piston, it is necessary in forging to carry these bosses right up to the head of the piston, and subsequently to machine away the interior so as to leave the bosses with a plane upper surface left by the boring tool. Such a piston is shown in Fig. 56. In cases in which extreme lightness is required, however, the boss is turned cylindrically by means of an overhung cutter carried on a bar passing through the gudgeon-pin hole and moved during the operation towards the piston shell. In consequence of the small diameter of the interior of the piston and the relatively large diameter of the outside of the boss, it is obvious that the overhung tool cannot be brought very close to the piston shell at the section through the gudgeon-pin axis normal to that of the piston. To enable

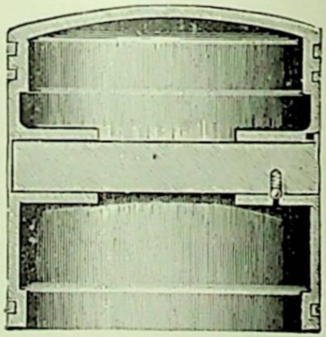


Fig. 57.—Section Welded Piston.

the cutter to remove as much material as possible, and to leave the boss with a fairly small radius at its junction with the piston, a pumping motor is sometimes employed on the boring bar to effect the removal of this small amount of metal; the extra amount of metal removed by this somewhat complicated operation, however, is very small.

In most steel pistons it will be found that the top is made appreciably thicker when of flat section than when it is convex or concave, which is due to the greater stiffness of the latter shapes. The thickness usually adopted for the tops of pistons from $3\frac{1}{2}$ to 5 inches in diameter is about $\frac{1}{8}$ inch; from 5 to $6\frac{1}{2}$ inches diameter it varies from $\frac{1}{8}$ to $\frac{3}{16}$ inch; and in pistons as large as 10 inches diameter it is increased to $\frac{3}{8}$ inch.

In many cases the diameter at the top of these pressed-steel pistons is made slightly smaller than that of the lower portion of the shell, for the same reason as in the case of cast-iron pistons, namely, to allow for the greater expansion of the hotter portion at the top. The thickness of the top of the piston is generally that of the greater portion of the body, and in particular is the same as the thickness at the bottom of the piston-ring grooves. For pistons from $3\frac{1}{2}$ to 5 inches diameter the depth of the groove for the piston rings usually amounts to about $\frac{3}{16}$ inch, and where a distance occurs between the lower piston ring and the top of the exterior of the gudgeon-pin boss, the thickness of the shell at that part is usually reduced to the ruling thickness. The thickness of the metal in the bosses is generally from $\frac{3}{16}$ to $\frac{1}{4}$ inch for the sizes just considered, whilst the combined length of the two bosses is from 40 to 55 per cent. of the piston diameter. The provision for securing the gudgeon pin in some cases takes the form of an auxiliary boss stamped on the lower face of the gudgeon-pin bosses toward the open end of the piston. In some cases these bosses are made thick enough to be screwed for receiving a set screw for locking the gudgeon pin; in certain special cases the bushes are made conical to secure conical locking bushes retaining the gudgeon pin in place.

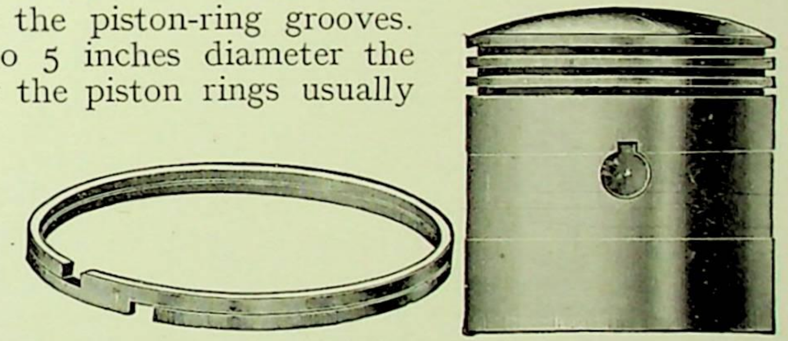


Fig. 58.—Welded Piston and Spring Ring.

It is usual in pressed-steel pistons to cut a few peripheral grooves in the skirt portion (below the gudgeon pin) for assisting the interception of the lubricating oil, but in some cases, as in the example shown in Fig. 57, there is provided an additional scraper ring at the lower portion.

Fig. 58 (right) is an outside view of a welded piston, whilst the illustration on the left shows a gas-tight double piston ring made by welding together (at the portion shown common to the two) the rings.

ALUMINIUM-ALLOY PISTONS

The use of aluminium-alloy pistons has increased progressively during the last few years, due to several reasons. Firstly, special alloys have been discovered which combine suitable strength with lightness and durability. Secondly, the earlier difficulties which greatly hampered the adoption of this type of piston, namely, that of excessive clearance between the piston and the cylinder, and caused hammering (or *piston slap*) when the engine was cold, has now been satisfactorily overcome in several ways, so as to allow for the greater expansion of the alloy. Thirdly, the difficulty in connection with the gudgeon-pin fixing has been overcome.

A typical aluminium alloy for pistons contains 92·5 per cent. aluminium, 4 per cent. copper, 2 per cent. nickel, and 1·5 per cent. manganese.

Aluminium-alloy pistons have been widely employed in aircraft engines and have given excellent results, and the experience gained has been found very useful in automobile-engine design. Before describing a few representative examples, a few words will be devoted to the principles involved in the use of aluminium alloys in cast-iron or steel cylinders.

In the first place, the specific gravity of the aluminium alloy used is about one-third that of cast iron, whilst its tensile strength is not very different, and in some instances is even greater. It is thus obvious that the aluminium alloy can be made much lighter for the same strength. Next in importance is the fact that aluminium has a heat conductivity which is about three times that of cast iron, so that the piston can be run much cooler. With cast iron the heat of the exploding and expanding gases raises the temperature of the piston crown considerably more, as the heat is not conducted away to the body and walls of the cylinder so quickly as in the case of aluminium alloy. It is thus possible, in the latter case, to run engines at higher speeds and compressions, from the point of view of lower weight and better thermal properties.

It is usual to obtain a weight reduction of from 25 to 50 per cent. over that of cast iron.

The principal difficulty encountered with aluminium alloys was due to the fact that, owing to the coefficient of linear expansion of the alloy being about three times that of cast iron, a relatively greater allowance had to be made in the diameter of the piston, so that when it was working at the usual engine temperatures it was not too tight. For this reason, also, the piston when cold was rather a slack fit in the cylinder; consequently, when warming up the engine there was appreciable piston slap.

The mean coefficient of expansion of the average aluminium alloy is ·000026, and for cast iron ·000010.

It is usual to allow from $\frac{4}{1000}$-inch to $\frac{6}{1000}$-inch clearance, between the piston and cylinder diameter, per inch diameter of cylinder for thermal expansion, and for another peculiar expansion which occurs and is termed *growth*.

In order to illustrate the method of arriving at the proper clearances to be allowed for aluminium-alloy pistons, consider an engine having cylinders of 3-inch bore; the necessary clearance in this case will be $3 \times \frac{5}{1000} = \frac{15}{1000}$ inch, so that the piston diameter will be 2·985 inch. In the case of aluminium pistons working in steel cylinders, there has been found, by experiment, to be a difference of temperature between the piston and the steel cylinder of 100° C., and a formula has been evolved for giving the relative expansion (or clearance) between the piston and cylinder as follows:

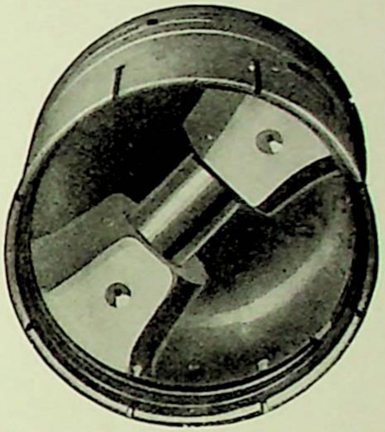


Fig. 59.—The Aerolite Aluminium-alloy Piston.

$$e = \frac{d\,(2{,}600 + 16t)}{1{,}000{,}000}\text{ inch}$$

where d = diam. of cylinder in inches
t = temp. of cylinder wall in °C.

The value of t, usually found by measurement of engine temperatures, varies from 130° to 170° C., and may be taken at 150° C. for general purposes.

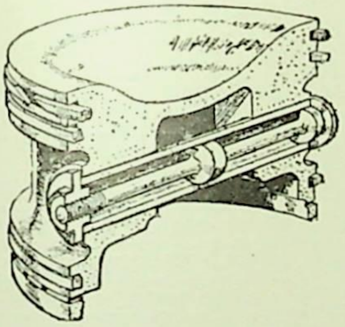


Fig. 60.—The Aluminium Piston employed on the Belsize-Bradshaw Engine (*Autocar*).

Fig. 59 illustrates a typical example of modern aluminium-piston design in the case of automobile engines, and shows also how the question of piston slap has been considered. The lower portion of the piston wall is split at intervals, and an internal-spring ring of steel is fitted near the lower edge, so as to ensure that the piston will fit the cylinder under all conditions.

Fig. 61 shows the Rudge motor-cycle piston, which is made in an aluminium alloy. This piston, it will be observed, has a domed top for the purpose of raising the compression of the engine to which it is fitted (in place of a flat-topped piston). It is of the waisted type, and immediately above the waist and also below it the surface is broken up into a number (30) of diagonal grooves. As the piston rises in the cylinder the oil is splashed upon the cylinder walls, and as it descends the bottom edge of the lower piston ring (of which there are three) scrapes down a wave of oil, which would otherwise offer a considerable fluid resistance were it not that the grooves provide an easy return of the surplus oil to the crankcase. These grooves maintain a uniform film of oil between the piston and the cylinder walls, but prevent an excess from passing the rings. The gudgeon pin in the case of this

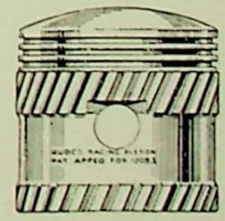


Fig. 61.—The Rudge Motor-cycle Piston.

piston is free in both the piston and the floating phosphor-bronze bush in the small end of the connecting rod ; it is retained in position by spring rings. The weight of this piston, which is 85 mm. diameter, is only 15 oz.

Fig. 62 illustrates the Ricardo slipper type of aluminium alloy piston used upon automobile and aircraft engines. It is so arranged that the two sides of the piston perpendicular to the axis of the gudgeon pin are cut away completely, and that the surface on the side of the piston which receives the connecting-rod thrust is of much larger area than the other side ; the weight of the piston is thus reduced by removing the bearing surface, which does little or no useful work.

This type of piston, owing to its smaller area in contact with the cylinder, gives a reduced frictional resistance, as the surface of oil which is being continuously sheared between the piston and the cylinder walls is appreciably reduced.

In the case of this piston a saving in weight of 50 per cent. over the usual cast-iron type is claimed ; owing to this weight reduction the reciprocating forces are reduced.

The results of a number of tests made with these aluminium pistons, when fitted to a light-car engine, showed that the brake h.p. was increased from 19 to 21·6, the lower figure being the maximum b.h.p. which was obtained when using cast-iron pistons. The slipper pistons weighed 0·64 lb. each, as compared with 1·37 lb. for the cast-iron pistons. The cylinder diameter in this case was $2\frac{3}{4}$ inches. The following table shows the relative weights of the three types of pistons referred to therein :

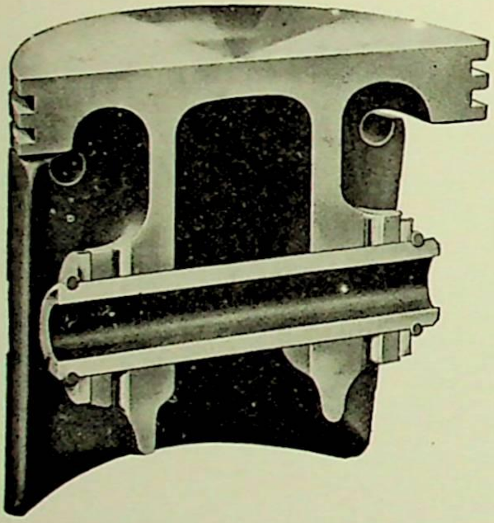


Fig. 62.—The Ricardo Piston in Section, showing Gudgeon Pin.

	Weights in lb.			
	Bare Piston. 4 inches diam.	With Rings, Gudgeon Pins, etc.	Total Weight of Reciprocating Parts.	Weight of Reciprocating Parts per sq. inch of Piston Area.
Aluminium Slipper Piston .	1·0	1·4	2·0	0·167
Aluminium Trunk Piston .	1·4	1·9	2·7	0·214
Light Cast-iron Piston . .	2·8	3·3	4·1	0·327

It is claimed that there is a reduction in the piston friction amounting to 30 to 35 per cent., and an increase in the maximum obtainable power of from 10 to 15 per cent., and about 5 per cent. at normal speeds. The oil consumption with this type is appreciably lower. In connection with the gudgeon-pin design for the slipper type of piston, it will be seen that a shorter and stiffer pin can be employed, so that there is a reduced tendency to bending.

The gudgeon pin is allowed to rotate freely in the connecting rod and in the bosses of the piston itself.

Owing to the design of this piston, with its cut-away side walls, the difficulties mentioned in connection with thermal expansion clearances are avoided and no piston slap occurs.

COMPOSITE PISTONS

Attempts have been made to combine the advantages of the lightness and heat-conducting properties of the aluminium piston with the wearing qualities of the cast-iron one. The principal objections which have been raised in the case of aluminium-alloy pistons are that the metal is comparatively soft, compared with cast iron, so that the piston grooves and the gudgeon-pin housing wear more rapidly. The special aluminium-copper alloys used are certainly a big improvement in this respect, but are still inferior to cast iron in the matter of wear. In the case of the Zephyr composite piston shown in Figs. 63 and 64 an aluminium-alloy head is provided to the piston, so that the heat is conducted away from the hottest part, namely, the piston crown, very efficiently. The skirt in this example is made of cast iron, so that there is no piston slap when the engine is warming up. The method of attachment of the skirt portion (which carries also the gudgeon-pin bosses) to the aluminium crown portion is by means of four studs and nuts.

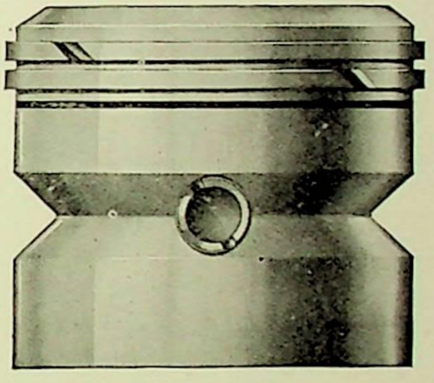


Fig. 63.—The Zephyr Aluminium Cast-iron Piston.

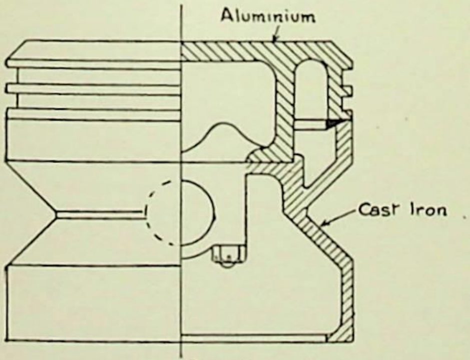


Fig. 64.

It is interesting to note that the Zephyr composite piston for the Ford engine weighs, complete with piston rings, only 32 oz. as against the 56 oz. for the standard Ford piston; a marked reduction in the reciprocating forces, and better acceleration and running are thus obtained.

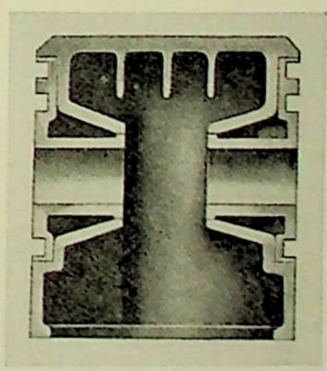


Fig. 65.—The Runbaken "Felco" Composite Piston.

Fig. 65 illustrates the Runbaken "Felco" aluminium piston, which is made by a process of casting a bronze insert into the aluminium shell.

The bronze portion carries the piston-ring grooves and the gudgeon-pin bosses, whilst the piston crown and skirt are made of aluminium alloy. This piston is about 35 to 45 per cent. lighter than a cast-iron one, and tests have shown that it wears equally well ; whilst the superior conductivity of the aluminium and bronze are beneficial factors in cooling the piston and in reducing the carbonisation.

METHODS OF SECURING GUDGEON PINS

A frequent source of cylinder scoring, in the case of early designs of petrol engines, was the gudgeon pin which had worked loose in the piston bosses. There are many methods which are employed to prevent this trouble, a few examples of which are illustrated in Fig. 66.

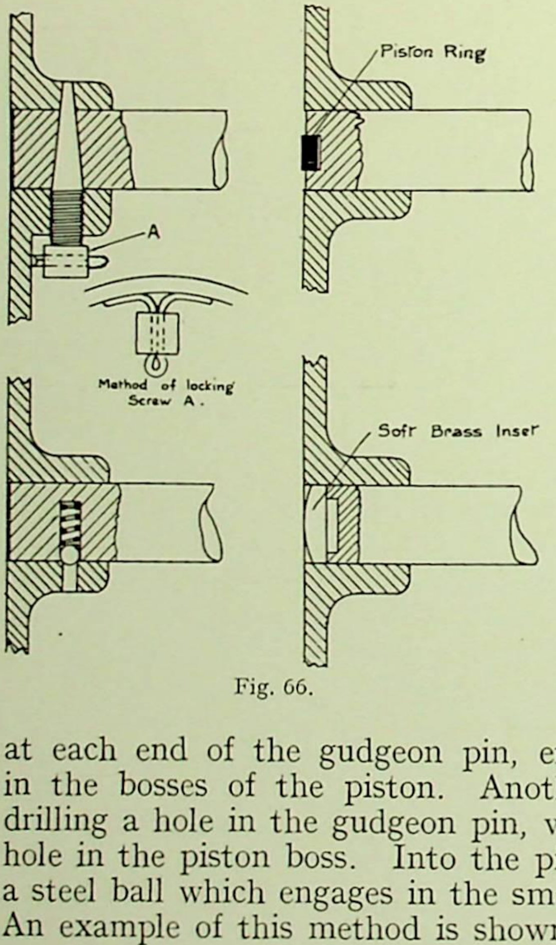


Fig. 66.

In the case of some well-known car engines no attempt is made to lock the gudgeon pin directly ; it is made a tight fit in the bosses and is shorter in length than the diameter of the piston by about $\frac{5}{16}$ to $\frac{3}{8}$ inch. At each end soft brass pieces are pushed into place, so that if the gudgeon pin works loose, no damage is done to the cylinder.[1] A piston ring is sometimes employed to hold the gudgeon pin, but cases have occurred in worn engines in which the piston ring has broken. The method used in the Rudge piston (shown in Fig. 61) consists in fitting spring rings at each end of the gudgeon pin, engaging in circular grooves turned in the bosses of the piston. Another interesting method consists in drilling a hole in the gudgeon pin, with a corresponding smaller size of hole in the piston boss. Into the pin hole is fitted a short spring with a steel ball which engages in the smaller hole drilled in the piston boss. An example of this method is shown in Fig. 66. The bearing pressure on the gudgeon pin should not exceed 1,500 lb. per sq. inch in normal types of engine ; the bearing area in this case is taken as the product of the length by the diameter of the bearing portion of the gudgeon pin.

PISTON RINGS

These are invariably made of cast iron. The usual method was to turn a number of concentric rings to a size a little larger than the cylinder bore, and then to saw, or machine out, a slot in each ring, afterwards compressing the ring so as to fit in the cylinder. It is obvious that this method results in a ring which only presses on the cylinder in two localities.

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[1] The method of clamping the gudgeon pin securely to the small end of the connecting rod, and arranging bearings in the piston bosses, is increasing in favour.

An advance on this method is to turn the inner surface of the ring eccentric with the outer, and to split the ring at the smallest section, as shown in Fig. 67 (B).

Piston rings are now frequently made as follows: the ring is first turned to a slightly larger diameter than the cylinder, and is then split and clamped with the split ends practically closed. The outer surface is next turned truly cylindrical and to a diameter a few thousandths of an inch smaller than the cylinder bore. On releasing the rings, they are found to fit the cylinders with an almost uniform radial pressure.

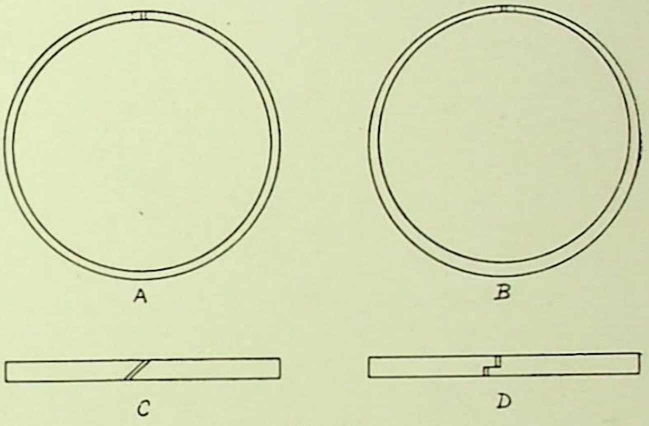

Fig. 67.

The two forms of slot most commonly employed with this type of piston ring are shown in Fig. 67; of these, that shown in D provides the more gas-tight joint and is made by the method of filing two slots half-way through the piston ring, one on each side, but each with one of its edges contiguous, so that the separation occurs at the adjacent corners of the two slots.

A type of piston ring widely employed in automobile practice is the hammered concentric ring. It is made of cast iron of a fine granular grey variety, and is uniform in thickness, so that it fills the piston grooves uniformly. A more or less uniform springing effect is obtained by hammering the inside of the ring with a ball-pointed hammer which leaves a series of shallow impressions readily observed in a new ring.

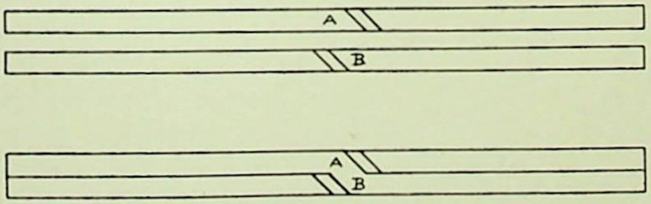

Fig. 68.—Showing Method of making a Compound, from two Simple Rings.

A number of composite or double-type piston rings are now in service, the object in each case being to provide better gas-tightness and more uniform radial pressure.

Fig. 68 illustrates one of the more common forms of double piston ring; that shown below is made by welding at one part two of the single diagonally slotted rings shown in Fig. 67 (C).

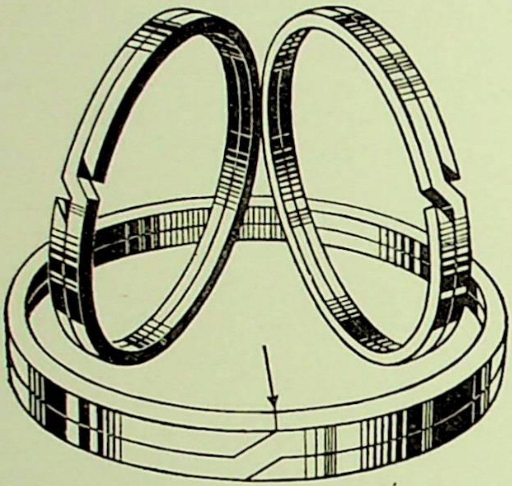

Fig. 69.—The Clupet Piston Ring.

A type of double ring, which is now being widely used, more particularly for replacement purposes, is the Clupet piston ring shown in Fig. 69. It will be observed that it provides an excellent non-leak obstruction to the cylinder gases, whilst giving the necessary spring pressure against the cylinder walls to ensure gas-tightness. It is recommended by the makers (Messrs. Clews, Petersen & Co., of London) that if substituted for the upper ring in ordinary cases, and for the two upper ones in the case of worn engines, a considerable improvement results; experience has amply justified these claims.

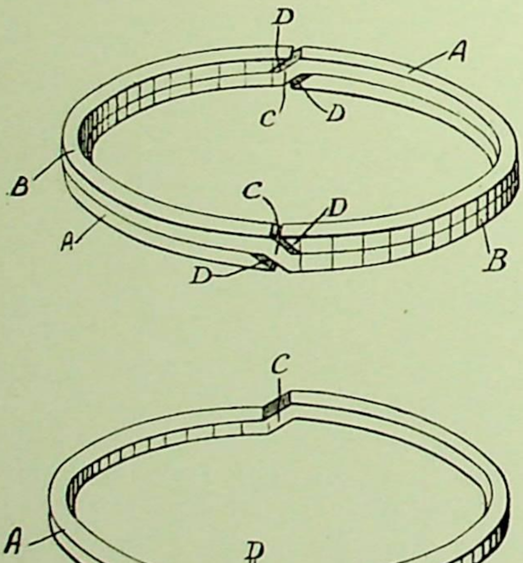

Fig. 70.—The Whidbourne Ring.

The Whidbourne ring shown in Fig. 70 consists of two ordinary piston rings machined in a special manner, as shown in the diagrams, so that when placed together they form a single ring of the same overall dimensions as one of the original rings. It will be observed that there are four gaps, and four free ends, so that an excellent and uniform bearing pressure is obtained, whilst there is no practical possibility of leakage of the gases. These rings can be milled out of ordinary rings, and to the same overall dimensions, and are economical to manufacture; they can also be used for replacement purposes, and with aluminium pistons.

Fig. 71 shows the McQuay-Norris "leak-proof" piston ring of American origin, the construction of which will be evident from the inset diagram. The "Superoyl" ring shown in Fig. 72 is intended for oily engines; the sharp diagonal edge is designed to catch up the excess oil off the cylinder walls and deposit it on the lowest part of the stroke at the base of the cylinder.

In the case of the eccentric type of piston ring, the " free " diameter, that is to say, the diameter of the open ring, removed from the cylinder, is usually made to 1·02 to 1·04 times the cylinder diameter. The greatest thickness of the eccentric type of ring is given by the relation—

$t = (\cdot 04d + \cdot 01)$ inch,

and the minimum thickness is 0·60 of the maximum thickness. The width of the groove is given by the relation—

$w = (\cdot 04d + \cdot 011)$ inch,

whilst the depth is usually taken as being 0·75 of the greatest thickness.

When the piston ring is compressed into the cylinder, the slots should have a clearance of from ·001 to ·003 inch; and the clearance of the rings in the piston slots, in the direction of the axis of the cylinder, should be from ·001 to ·002 inch.

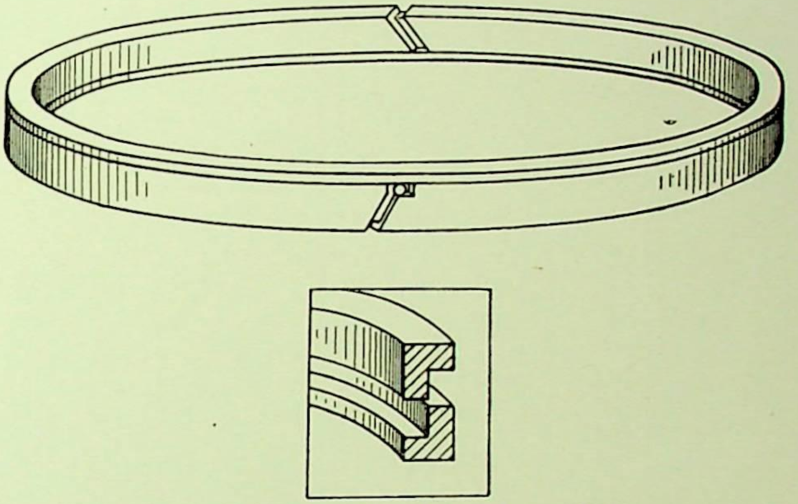
Fig. 71.—The McQuay-Norris Piston Ring.

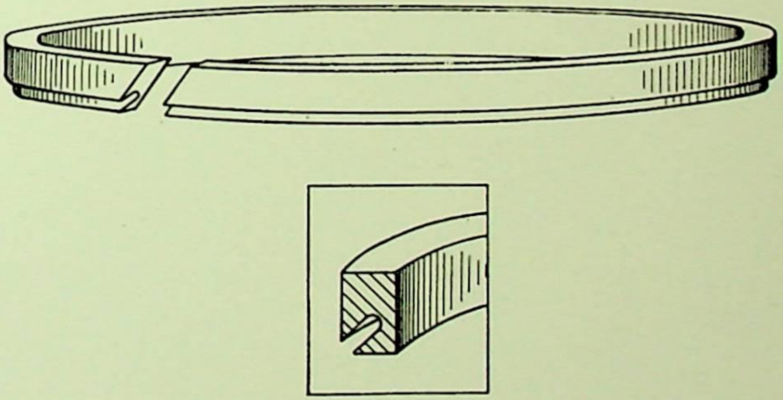
Fig. 72.—The "Superoyl" Piston Ring.

CONNECTING RODS

The connecting rod transmits the load on the piston to the crank, and it is usually under a compressive force. The greatest load which can occur[1] is due to the explosion pressure, and may be taken as being equal to the product of the explosion pressure by the area of the piston. Thus, in the case of a $3\frac{1}{2}$-inch piston, and for a compression ratio of 5, the load on the piston during explosion will amount to about 3,700 lb., that is, about 1·6 tons. The connecting rod must therefore be exceedingly strong as a strut, and, for reasons which we cannot enter into here, it must be sufficiently strong to resist a side-bending action which occurs. The usual shape of connecting-rod section is the I beam or H section as shown in Fig. 73, although tubular rods are sometimes employed. Connecting rods are now invariably made as drop-forgings in nickel-chrome or chrome-vanadium steel, heat treated after machining to the final proportions, so as to give a breaking strength in tension of between 60 and 80 tons per sq. inch.

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[1] In ordinary automobile engines, and disregarding piston inertia effects.

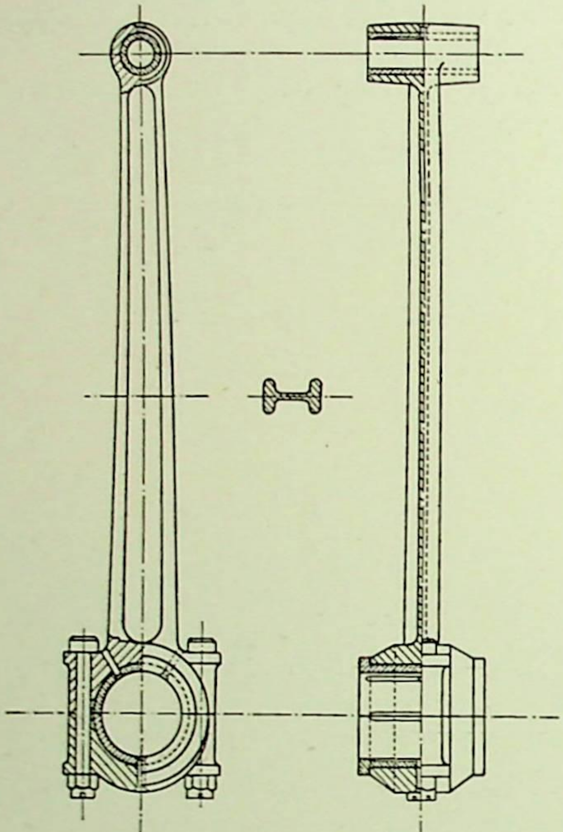


Fig. 73.—A Typical Petrol-engine Connecting Rod.

Fig. 74 shows a typical design of automobile-engine connecting rod, and illustrates the method of providing the necessary bearings for the gudgeon-pin and crank-pin journal. It is usual to provide white-metal-lined phosphor-bronze or gun-metal bearings for the crankpin journals, the two bearing shells being pinned to prevent their rotation in the connecting rod and its cap.

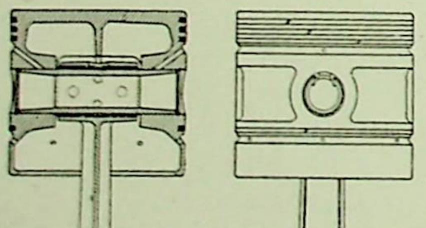
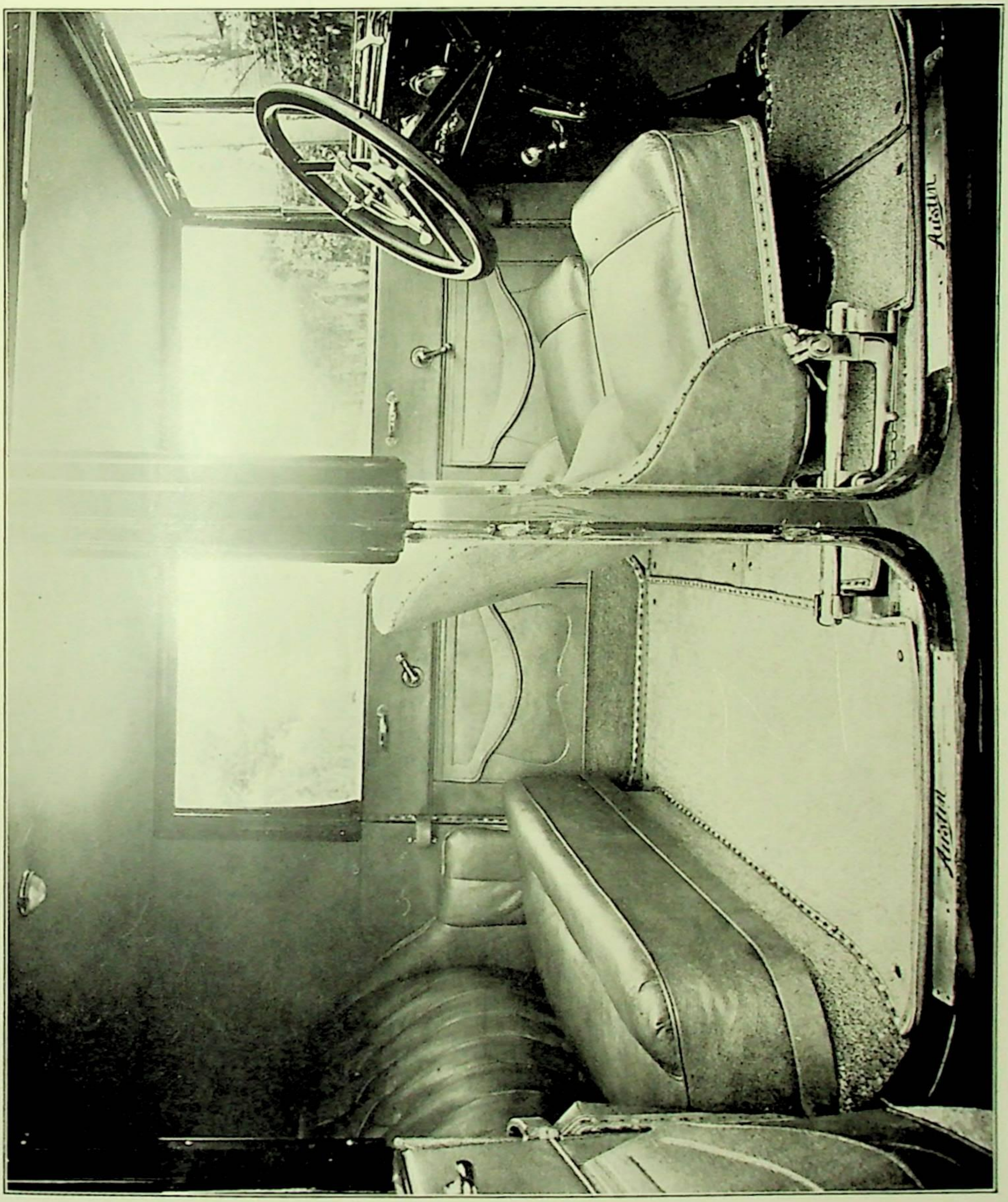


Fig. 74.—High-speed Engine Aluminium Piston (with Oil-scraping Grooves), and Connecting Rod.

The white-metal bearings are now die-cast to within one or two thousandths of an inch of their final diameter, and several thin packing strips or "shims" are fitted between the two halves of the complete bearing before it is machined to fit the crankpin; in this manner a small amount of wear in the bearing metal can be taken up by removing one or more of the shims.

The connecting-rod cap is bolted to the main rod by two strong high-tensile steel bolts, and great care is taken in the locking of the nuts usually by split pins, spring washers, and lock nuts.

The bearings themselves are provided



Interior View of 20 H.P. Austin Limousine.

with oil grooves, radiating from the oil holes, and the latter are fed by oil, either under pressure through holes in the crankpin from an oil-pressure system, or by means of oil-scoops formed on the lower cap of the big-end bearing. These scoops at the lowest portion of their movement dip into oil troughs in the crankcase, and pick up sufficient oil to lubricate the bearings, the oil grooves in which can be plainly seen in the illustrations. The connecting rod, for theoretical reasons, should be fairly long; but for practical reasons connected with the over-all height of the engine, it is usually made about $1\frac{3}{4}$ to $2\frac{1}{2}$ times the piston-stroke. The approximate proportions of the section of an H-shaped rod are as follows, namely: *Depth*, from $\frac{1}{3}$ to $\frac{1}{4}$ the cylinder bore; *width*, from $\frac{1}{7}$ to $\frac{1}{9}$ the cylinder bore; and *thickness* of webs about $\frac{1}{25}$ to $\frac{1}{30}$ the cylinder bore.

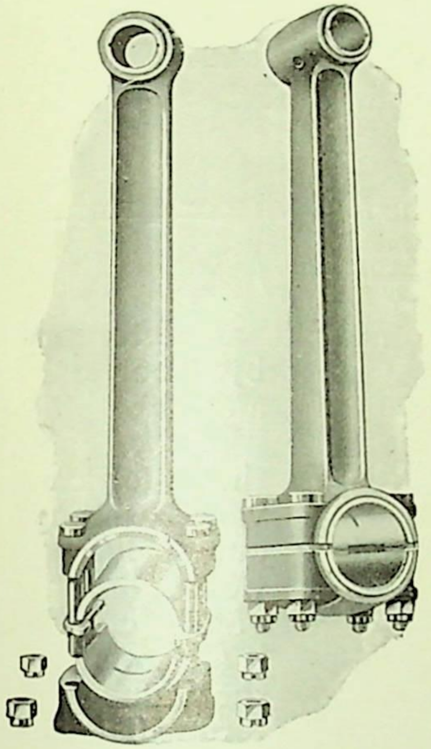


Fig. 75.

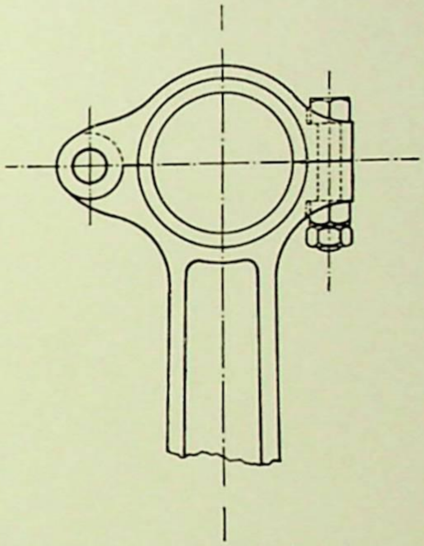


Fig. 76.—Showing Method of clamping Gudgeon Pin to Small End of Connecting Rod.

In some cases the small-end phosphor-bronze bush in the connecting rod does not rock on the gudgeon pin fixed in the piston bosses, but the bush is dispensed with, and the gudgeon pin is clamped in the split small end of the connecting rod, and rotates in bushes fitted in the piston bosses; this practice, which is adopted in the case of the Ford engine, has certain advantages from the manufacturer's point of view, but gives a heavier rod than in the other case.

THE CRANKSHAFT

The design of the crankshaft is dependent upon the number and arrangement of the engine cylinders, and upon the mode of construction adopted.

Before considering the various designs of crankshaft in general use, reference will be made to the theoretical aspect of the subject, in order that a working idea may be arrived at of the forces which the crankshaft must withstand.

Consider the line diagram shown in Fig. 77, in which P represents the piston, PC the connecting rod, and CO the crank arm, O being the axis of the crankshaft.

The force P on the piston produces a thrust $Q = \frac{P}{\cos\phi}$ in the connecting rod, and this causes at any moment a turning effort or "torque," $T = Q.Om = \frac{P.Om}{\cos\phi}$.

This expression can be shown to be equal to the following one, namely:

$$T = \frac{P.Om}{\cos\phi} = P.r\sin\theta\left(1 + \frac{r\cos\theta}{\sqrt{l^2 - r^2\sin^2\theta}}\right),$$

which gives the value of the torque at any moment as determined by the crank-angle θ.

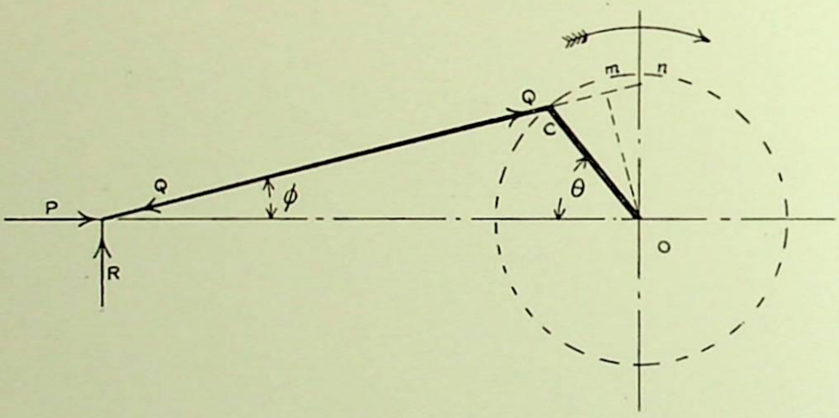


Fig. 77.

Now, since the piston load P varies throughout a cycle, as well as the angle of the crank θ, it follows that the value of the torque T is constantly changing throughout the working cycle. As would probably be anticipated from an examination of the pressure variation (or indicator) diagram, the piston pressure is a maximum near the beginning of the stroke and varies very considerably more than the expression $\frac{Om}{\cos\phi}$, so that the greatest torque may be predicted as occurring near the beginning of the explosion stroke.

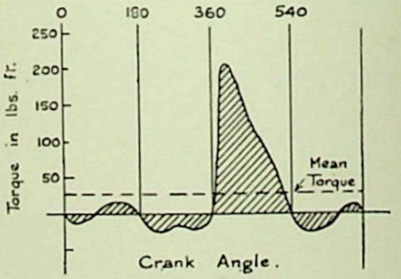


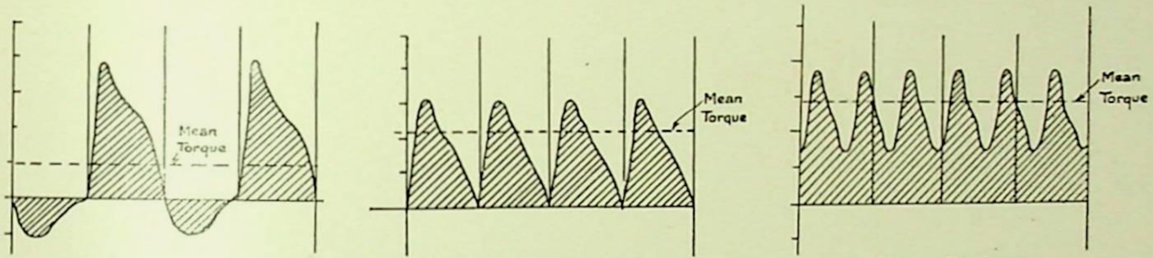
Fig. 78.

If the values T of the above expression be worked out, by using a diagram of cylinder pressures for P, and by graphically (or otherwise) finding ϕ and the distance Om, then the results may be conveniently plotted on a crank-angle base, and studied with ease.

An example of such a diagram is shown in Fig. 78, and is drawn for the case of an engine of 3 inches bore and 4 inches stroke, the ratio of the connecting rod to the crank-arm length being 4. The diagram has, however, been corrected for the inertia effect, due to the fact that a force is required to accelerate the piston and part of the weight of the connecting rod at the beginning of the stroke, but the energy of this force is recovered during the retardation or stopping of the piston, etc., at the inner end of its stroke. It is not important in slow-speed engines.

An examination of the torque diagram in Fig. 78 shows at once that there is a very considerable fluctuation in the turning effort on the crankshaft ; in effect, the effort resembles a jerk during the explosion stroke. It is therefore necessary to provide a flywheel of appreciable weight, or rather, moment of inertia, in the case of a single-cylinder engine, in order to store up part of the energy of the explosion-effort, and to use this energy to carry the mechanism over the exhaust, inlet, and compression periods.

Consider for a moment how this maximum torque compares with the mean torque throughout a cycle. The value of the mean torque can be estimated as follows :



Two-cylinder opposed. Four-cylinder. Six-cylinder.

Fig. 79.

$$\text{Mean torque} = 5252 \cdot \frac{\text{b.h.p.}}{\text{r.p.m.}} \text{ lb. ft.}$$
$$= \frac{\text{Brake Mean Pressure} \times A.l.}{4\pi \times 33{,}000} \text{ lb. ft.}$$

where b.h.p. is the brake horse-power, r.p.m. the revolutions per minute, A the area of the piston in sq. inches, and l the stroke in feet. In the present example the ratio of the maximum to the mean value of the torque is 8·6, the mean torque being shown to scale by the dotted line annotated.

It is quite possible that many of the earlier fractures of shafts and parts driven from the crankshaft were due to the designer omitting to take into account the maximum value of the torque, instead of which the average value was assumed.

It also follows that as the number of cylinders is increased in a given engine, so does the value of the ratio of the maximum to mean torque become less,. and therefore the crankshaft, other factors remaining the same, can be made appreciably lighter for a multi-cylinder engine than for a single-cylinder one.

Fig. 79 illustrates the torque diagrams for a two-cylinder opposed, a four-cylinder vertical, and a six-cylinder vertical engine. The ratios of the maximum to the mean turning efforts, are 3·9, 2·0, and 1·4 respectively. The advantages of even torque will be evident from these results.

Coming back once more to the practical side of the subject, and considering the materials used for, and the methods of manufacture of, crankshafts, these have now settled down almost to fixed standards.

The materials employed for petrol-engine crankshafts are now invariably the high-tensile alloy steels, such as chrome-vanadium steel or nickel-chrome steel. The shafts are heat-treated after rough machining and the journals are ground to final sizes after. The tensile strengths of the steels employed vary from 60 to 80 tons per sq. inch, and the hardness from 250 to 330 on the Brinell scale (i.e. extremely hard).

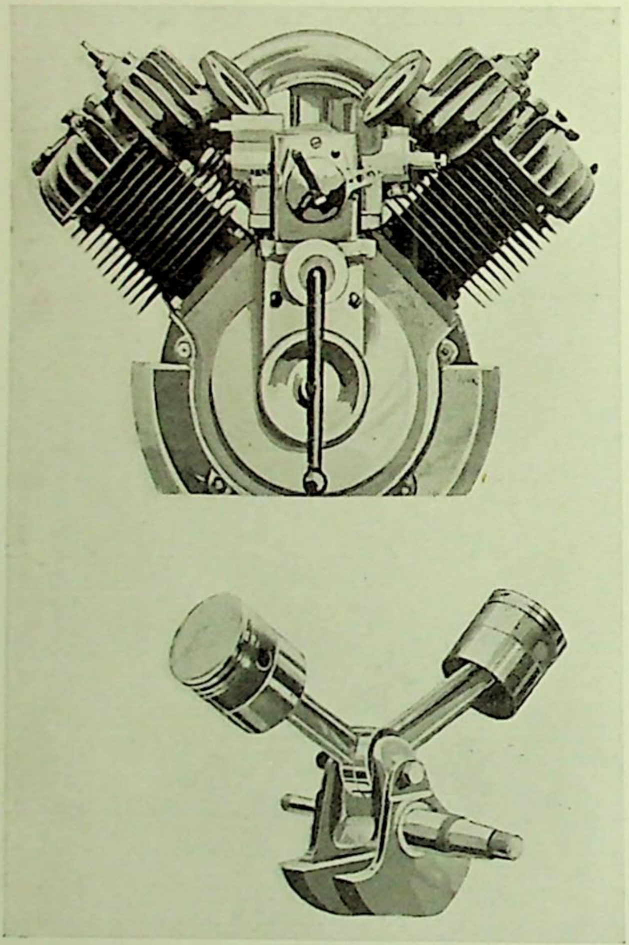


Fig. 80.—The G.N. Engine and Crankshaft.

Crankshafts are made by one of three principal methods as follows :

(1) Built up from components, as in motor-cycle engines.

(2) Rough forged and machined from the solid.

(3) Drop forged to the final shape, but not to final dimensions.

Dealing with these methods in the order named, the built-up crankshaft is usually a convenient combination of the main driving shaft, the flywheels with their balancing weights and the crankpin. Fig. 106 illustrates a motor-cycle type of crankshaft unit, whilst the crankshaft for the G.N. twin-cylinder Vee-type engine is depicted in Fig. 80, together with the pistons and connecting rods.

In the former case, the pins or shafts are fastened to the flywheels by means of the tapered portions and nuts shown, there being keys for the main shaft taper portions to take the turning moment. In order to ensure accuracy of alignment on reassembly, as for example after fitting new crankpins or bearings, it is now usual to provide a pair of holes directly opposite one another in the opposite side of the flywheel to the crankpin. By introducing a parallel rod of the correct size into these holes the two flywheels can be accurately aligned, and the crankpin nuts tightened up and locked. In the case of the Triumph engine the gudgeon pin was also designed for this purpose.

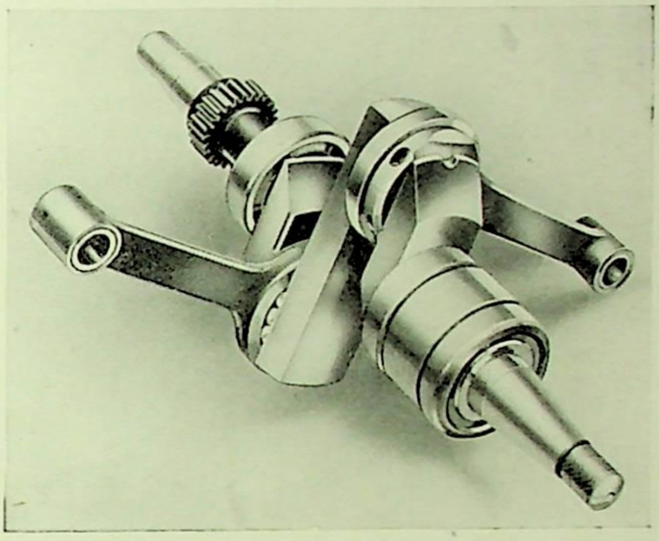


Fig. 81 —The A.B.C. Engine Crankshaft and Connecting Rods, with Roller-bearing Big Ends.

Another example of a built-up crankshaft, in the case of a four-cylinder engine, having three bearings of the Skefko ball-bearing type, is shown in Fig. 82. In order to get the middle ball races into position, it was necessary to make the crankshaft in two portions, the one being provided with a tapered end, to fit in a corresponding tapered hole in the other, immediately under the centre bearing; this method provides a stiff crankshaft. The diagram shows also the manner in which the main bearings are carried from the upper portion of the crankcase casting, and also the attachments of the flywheel and the starting dog-pinion on the left-hand side.

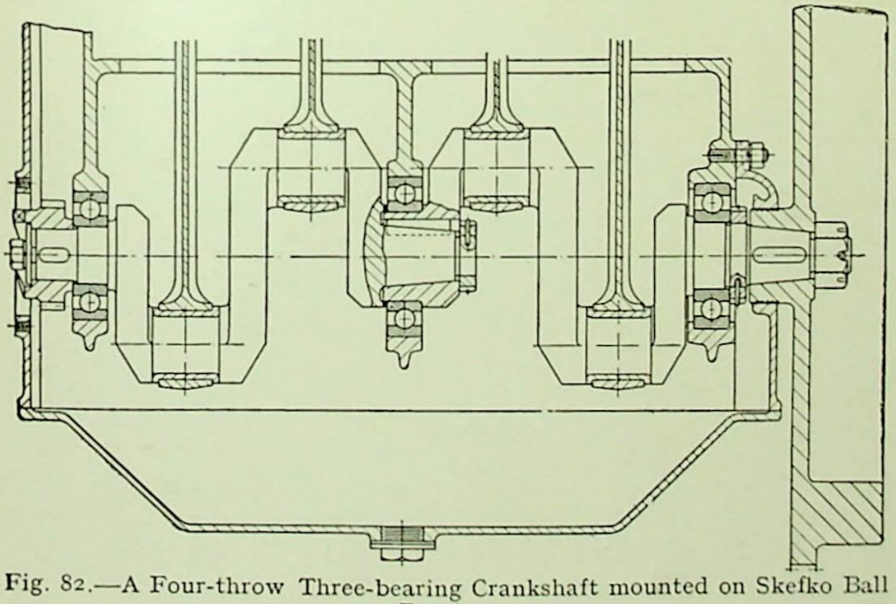
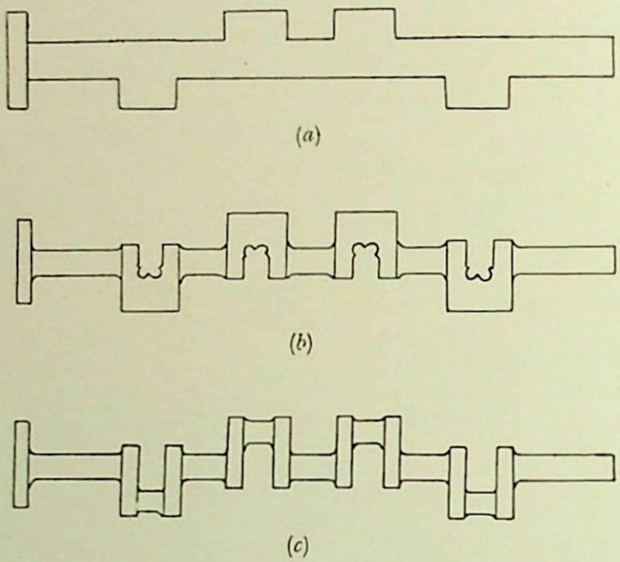


Fig. 82.—A Four-throw Three-bearing Crankshaft mounted on Skefko Ball Bearings.

The second method of making a crankshaft is illustrated in the line diagrams shown in Fig. 83 (*a*), (*b*), and (*c*). The billet from which the shaft is made is a flat forging, and the first machining operation consists in drilling the holes shown, after which the portions of metal not required are removed, and finally the shaft is machined in the usual manner, heat-treated, and then ground to the required dimensions, The third method is the one which is the most universal to-day, as the modern drop-forging press can cope with practically any size of petrol-engine shaft and give a finished product requiring only machining along the bearing and journal parts.

Fig. 84 illustrates a drop forging of a six-cylinder engine crankshaft made by Messrs. Vickers, Ltd. The method employed consists in using metal dies of the correct forms to give the final shape required. The rough-forged red-hot metal is placed on the lower die, which is affixed to the anvil or table of the hydraulic press; the upper die is next placed in position and pressure applied by means of the press, or automatic hammer used, and the hot metal flows to the shape of the dies, leaving only thin metal extrusions corresponding to the junctions of the dies. These extrusions or "fins" are usually removed by means of trimming dies, so as to leave a much better final shape.


(a)

(b)

(c)

Fig. 83.

Obviously, the cost of the metal dies for the types of crankshaft used in motor-car practice is considerable, and drop forging is only commercially possible where minimum quantities of 100 or so are required. Drop forgings possess the great advantage over straight-drawn and forged parts that the grain of the metal can be made to flow in the directions most advantageous from the point of view of strength. Thus in the case of a gear blank the grain can be obtained radially

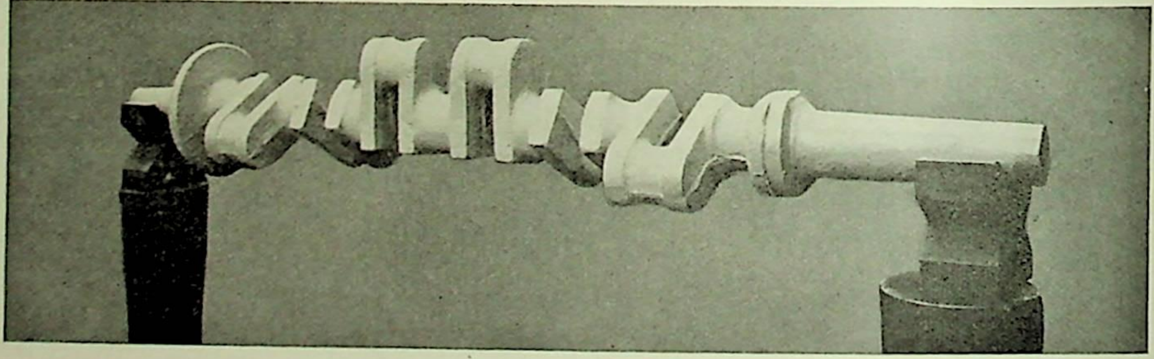


Fig. 84.—A Six-cylinder Engine Crankshaft Drop Forging.

instead of parallel to the axis, as in the case of a gear cut from a solid bar. Similarly with crankshafts, the grain on drop forgings follows the general direction of the sides of the webs and the axes of the journals and bearing portions.

Another advantage of drop forgings is their uniformity in dimensions, and the accuracy to which they can be made, so that the machining operations are reduced to a minimum. It is possible to drop forge to an accuracy of about $\frac{1}{32}$ inch, so that the only metal to be taken off is on the bearing portions. An excellent illustration of the utility of the drop forging from this viewpoint is in the case of the Dorman 12-h.p. engine crankshaft, in which the crank-web balance weights are drop forged in position as shown in Fig. 85.

It is necessary to provide these balance weights for high-speed crankshafts, for although in the case of a four-throw crankshaft the two outer cranks balance the two inner, yet there is a double bending effect on each of the two outer pairs of cranks which tends at high speeds to flex the shaft, that is, to cause it to whip. By arranging balance weights of the correct weight and at the correct radii, so that the centrifugal forces on the balance weights counteract those on the cranks themselves, this effect is nullified, and high speeds can be attained without vibration and risk of breakdown. Moreover, the main bearings will not tend to rock or "tip," in the case of balanced crankshafts, so that less friction will result. In the example illustrated the front journal runs in a large-diameter self-aligning ball bearing, and at the rear in a very long plain bearing. Against the latter is a steel double-thrust washer which relieves the ball bearing at the front of all end thrust. The rear bearing is mounted in a detachable end plate on the crank chamber, the opening which the plate covers being large enough to permit the crankshaft to be threaded through it in course of erection.

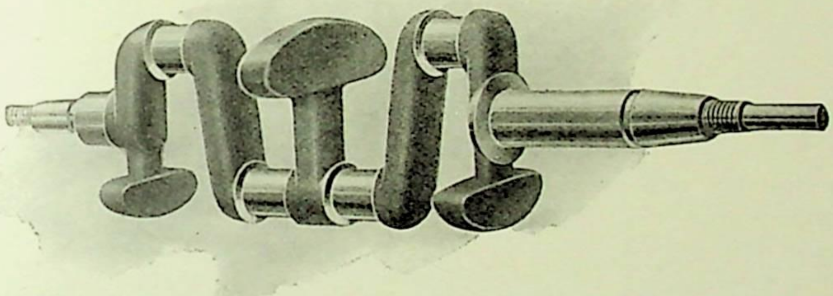


Fig. 85.—The 12 h.p. Dorman Crankshaft.

An example of a four-throw type of crankshaft machined all over is given in Fig. 97. In this case there are three main bearings of the white-metal-lined phosphor-bronze shell type. It is not common practice nowadays to provide a bearing between each crank, as this leads to a heavier and somewhat longer engine, and is not compatible with the use of monobloc cylinders. If anything, the tendency is toward the two-bearing type of crankshaft shown in the previous illustration (Fig. 85), and also in the sectional diagram in Fig. 86, which depicts the application of Skefko self-aligning bearings to a petrol-engine crankshaft of the four-throw type.

A good example of a finished six-throw crankshaft is given in the case of the Lanchester six-cylinder 40 h.p. engine ; Fig. 87 illustrates the general design of this crankshaft.

It is provided with a main bearing between each pair of cranks

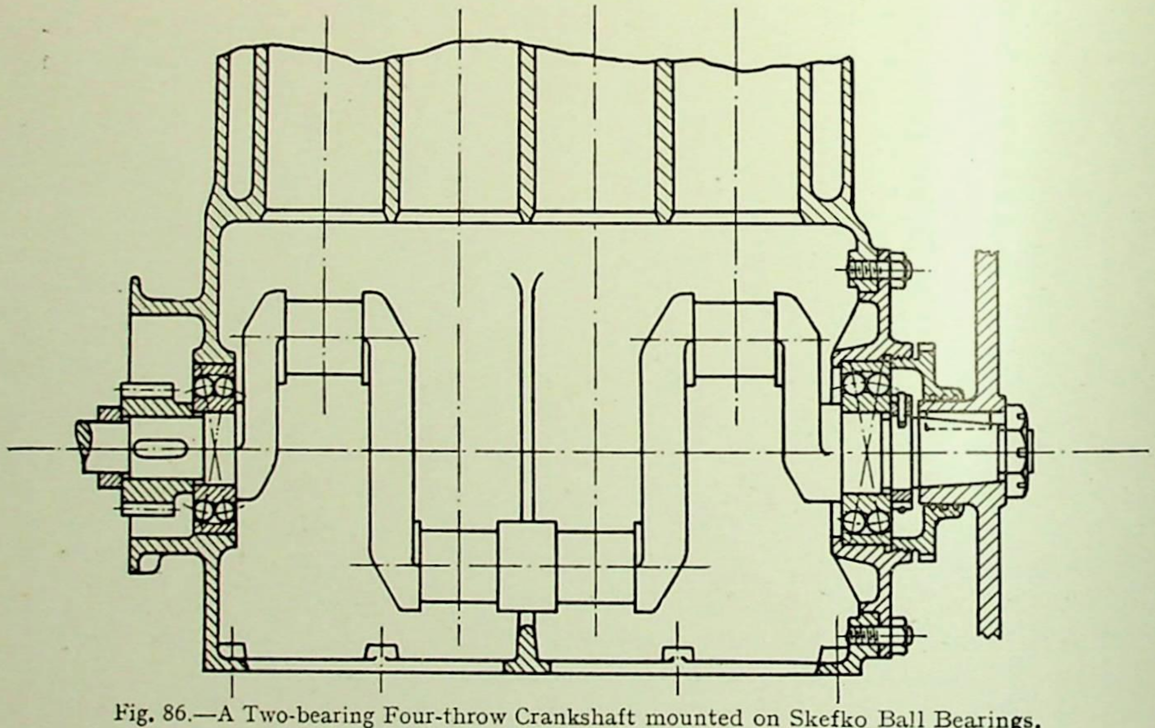


Fig. 86.—A Two-bearing Four-throw Crankshaft mounted on Skefko Ball Bearings.

and at each end, and is thus exceedingly rigid. The crank is of hollow formation, the shaft and pins being bored to effect a saving in weight ; the holes in the crank are plugged at the ends, and the oil-pressure system of lubrication is employed. At the left-hand end the starting dog which engages with corresponding teeth in the starting handle can be seen, and behind it is a worm gear for driving the worm wheel at the lower end of the vertical shaft for driving the overhead camshaft. The rear gear wheel is for driving the layshaft ; which in its turn operates the oil pump and dynamo. This layshaft is also driven by the electric-starter motor, the starter gears being arranged so that they come to rest and do not revolve immediately the engine is started.

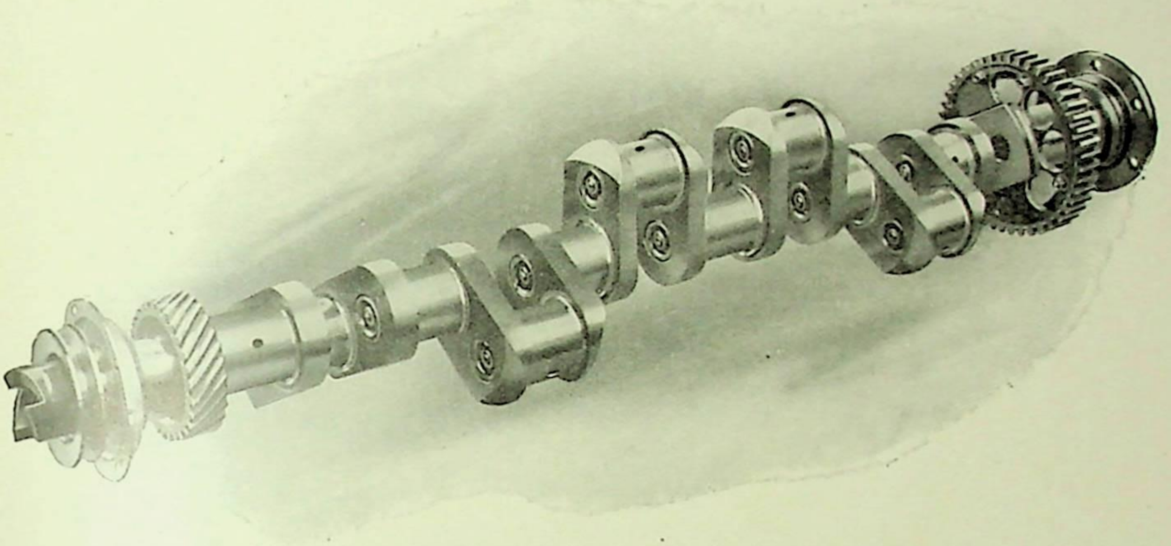


Fig. 87.—The Lanchester Crankshaft.

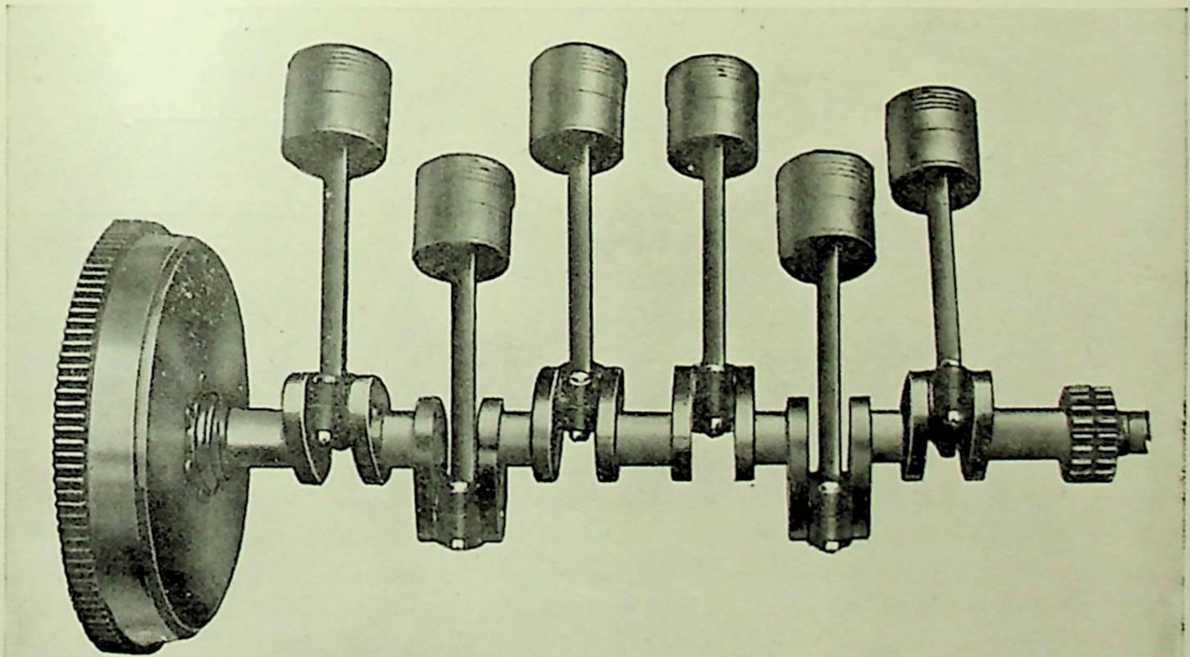


Fig. 88.—The Packard Six-cylinder Engine Crankshaft.

The crankshaft for the Packard six-cylinder engine, with the connecting rods, pistons, and flywheel, are shown in Fig. 88. This crankshaft is of the seven-bearing type, so that it is very rigid in action.

As regards the dimensions of the crankshaft diameters, these are carefully calculated from the turning-moment diagrams, and for the particular type of steel employed; but experience in the matter of crankshaft design, in connection with stiffness and the absence of "periods" or vibrations, is a valuable asset in the final assigning of dimensions. It is usual to make the width of the crank $w = 1{\cdot}35$ (the diameter of the crank journal). The thickness of the webs, $t = 0{\cdot}65\, d$.

An approximate relation between the crankshaft diameter d and the cylinder diameter D is as follows:

$$d^3 = 0{\cdot}5\, D^2$$

and for the crankpin diameter—

$$d_1 = 1{\cdot}05\, d;$$

The length l of the crankpin is given by—

$$l = 1{\cdot}3 d_1;$$

The proportions of the crankpin and journals are usually fixed finally from considerations of the allowable bearing pressure, and the product $l \times d_1$ is taken as the projected bearing area.

If P is the maximum load on the crankpin, then the quantity $\frac{P}{l \times d_1}$ should not exceed 700 lb. per sq. inch.

In the case of the main bearings the bearing pressure should not exceed 400 lb. per sq. inch in the case of high-speed engines, and 500 to 600 lb. per sq. inch for moderate speeds.

The main bearing nearest the flywheel has to carry the weight of the flywheel in addition to the working loads, and for this reason is made appreciably longer than the other bearings. It is usually made about 50–100 per cent. longer than the intermediate bearings. In connection with the length of the crank-pin journals, this has to be considered in relation to the overall length of the engine, and for a compact engine it is desirable to increase the diameter of the crankpin rather than to increase its length in order to obtain the required bearing area. On the other hand, the crankpin diameter must not be too large, or the rubbing velocity will be too high and what is gained in bearing area will then be offset by the increased rubbing velocity.

It should here be explained that the product of the bearing pressure by the rubbing velocity is the ultimate factor which decides the proportions of a bearing; rubbing velocities for crankpins and main bearings may be taken as being of the order of 17 to 25 feet per second for modern engines.

THE CRANKCASE

The crankcase performs the functions of housing the crankshaft and camshaft bearings, of enclosing the lubrication system, and of securing the whole engine, including the cylinders bolted to it, to the frame of the chassis.

Crankcases are now either cast integral with the cylinder unit, or are made so that the cylinders bolt down directly on the top machined face.

Externally, the complete engine consists usually of the detachable cylinder head and water cover-plate, the cylinder unit, the crankcase unit, and the oil sump; the latter may be integral with, or separately attached to, the crankcase.

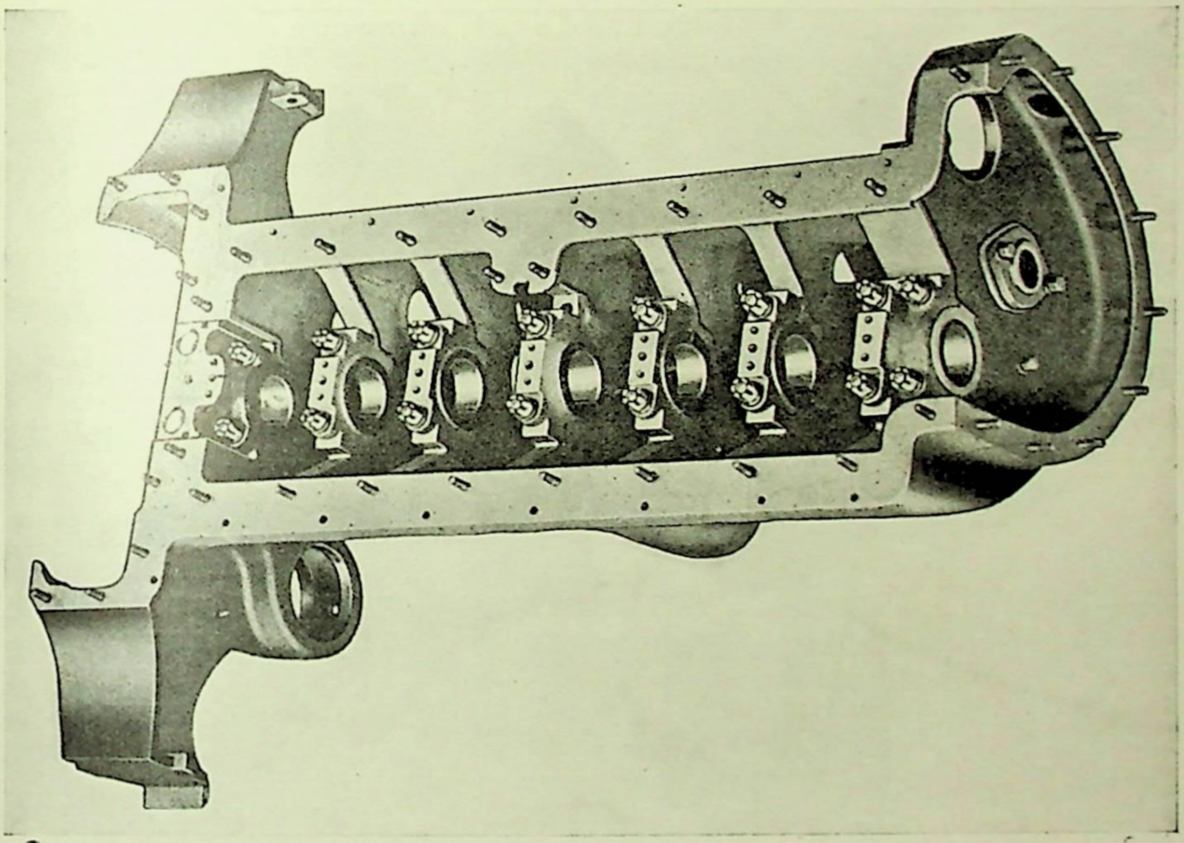


Fig. 89.—The Packard Crankcase (Lower Half).

Crankcases are generally cast in an aluminium alloy containing about 84·5 per cent. aluminium, about 13 per cent. of zinc, and 2·5 per cent. of copper, and are made in one unit, as shown in Fig. 27, with the housings for the crankshaft and camshaft bearings, the machining surfaces for the timing-case cover, oil sump, cylinder unit, flywheel or clutch casing, and the holding-down webs, all cast in the correct positions. The various machining operations are then carried out in jigs, to ensure accuracy of fitting of the parts which are machined separately and afterwards bolted to the crankcase. A typical example of the lower half of a crankcase, for a seven-bearing six-cylinder engine, is that of the Packard engine shown in Fig. 89.

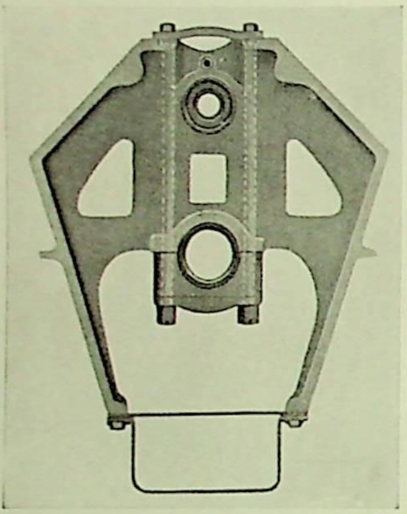


Fig. 90.

Fig. 90 illustrates a cross-section through the crankcase of the Cadillac eight-cylinder Vee-engine, and shows the long bolts employed for holding the main-bearing caps in position, also the overhead camshaft bearing.

The arrangements of the crankcase bearings and the oil sump are dealt with in the section on lubrication, and illustrations are there given of several alternative designs.

In the case of certain engines, chiefly of American origin, the cylinders and upper half of the crankcase are made in one casting in iron, whilst the lower half of the crankcase is made either as a casting, containing the oil sump, or in the form of a sheet-steel pressing. The aluminium casting is, of course, lighter, and for this reason is employed more universally in engines made in this country. The Continental 13·2 h.p. engine shown in Fig. 30 is an example of this type.

FLYWHEELS

These are fitted on the side remote from the radiator, in order to store up part of the energy given out by the engine during the firing stroke, so as to carry the crankshaft around during the ensuing exhaust, induction, and compression strokes. The flywheel also serves the secondary purpose of housing the clutch unit, and in some cases has a series of peripheral teeth cut in its surface for the pinion of the starting electric motor to engage with. Examples of these are given in the section dealing with clutches.

Obviously, the proportions of the flywheel for a single-cylinder engine will be relatively greater than for a two- or four-cylinder one, and it will be found that as the number of cylinders in an engine unit increases, so does the size of the flywheel decrease. This is due to the fact that the torque becomes progressively more uniform as the number of cylinders increases, as has already been shown in the torque diagram.

In the case of motor-cycle engines the flywheel is divided into two parts, and the crankpin serves as a connecting link between the two, as will be seen from Fig. 106.

For single-cylinder engines the weights of the flywheels usually work out at about 1 to 2 lb. per cubic inch of cylinder capacity, whilst for four-cylinder engines they vary from $\frac{1}{2}$ to $1\frac{1}{2}$ lb. per cubic inch.

It should be explained here that it is not the actual weight of the flywheel, but the distance from the axis at which this weight is disposed, that matters in connection with the storing up of energy. The moment of inertia, upon which this latter property depends, is proportional to the weight or mass, and to the square of the distance of this mass from the centre, or axis.

Thus by giving as large a diameter as possible, and by concentrating most of the weight in the rim of the flywheel, the greatest moment of inertia, and therefore the greatest energy-storing capacity, will be obtained. The smooth running of certain single-cylinder engines with relatively large diameter outside flywheels is almost entirely due to the above factor.

Flywheels are usually made in cast iron, or in good medium carbon steel.

THE EXHAUST SYSTEM

The exhaust gases are taken from the engine through a steel or copper pipe, or pipes, to a silencing chamber, from which they are ejected into the atmosphere.

It is usual in the case of single or two-cylinder engines to employ one or two pipes respectively, to which steel flanges are welded, direct from the exhaust port outlet on the cylinder to the silencer. In the case of four-cylinder engines, it is now usual to cast, in iron, the exhaust manifold with four carefully designed sweeps leading into a common cylindrical chamber, flanged at one end; the exhaust pipe then bolts on to this flange and leads the gases to the silencer, as shown in Fig. 91. The exhaust manifold is often held firmly to the cylinder by means of studs and nuts, or alternatively a smaller number of studs (one per cylinder) is used, but with Tee pieces which hold down in two places, instead of one.

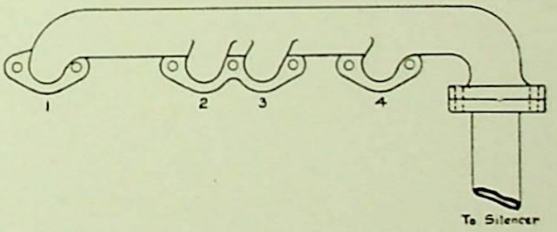


Fig. 91.

Fig. 92 shows the 10 h.p. Dorman engine combined inlet and exhaust branch, which possesses the advantage that the heat from the exhaust portion materially assists in vaporising the fuel before it enters the

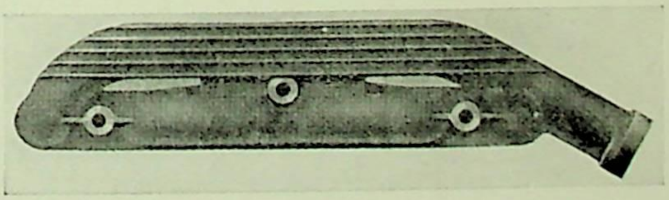


Fig. 92.—The Combined Inlet and Exhaust Branch of the 10 h.p. Dorman Engine which appreciably assists in vaporising the Fuel.

engine; this method of utilising the heat of the exhaust is known as the *hot-spot* system. It is usual in hot-spot systems to design sharp bends or projections in the inlet system which are heated more or less locally by the exhaust gases, but unless carefully designed a loss of charge or volumetric efficiency results.

The internal diameter of the exhaust pipe is commonly made about one-third to one-half of the cylinder diameter for a four-cylinder engine. The function of the silencer is to reduce the noise of the exhaust with the least possible loss of power. The silencer should allow the exhaust gases, which enter it in a series of high-velocity puffs, to issue in an almost continuous stream, with little intermittency of pressure, and therefore little acoustical effect.

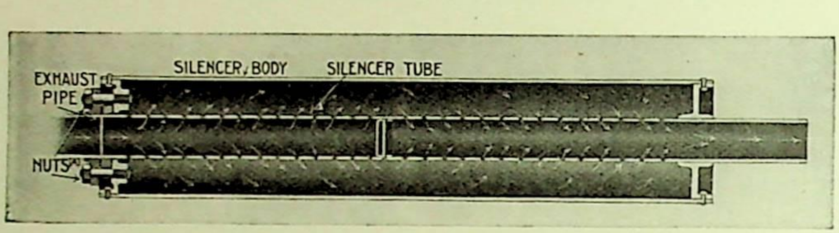


Fig. 93.—The Wolseley 10 h.p. Engine Silencer.

The silencer usually employed upon motor cars consists of a cylindrical sheet-metal box closed at the one end except for the exhaust inlet pipe, and with an exhaust outlet pipe at the other. A series of plates drilled with small holes, which serve to baffle or to break up the main puffs, is usually fitted in this box.

Fig. 93 illustrates the silencer design in the case of the 10 h.p. Wolseley engine, the paths of the gases being indicated by the arrows. It is essential to be able to clear a silencer of the soot which accumulates after long periods of running, and the design should render this operation a simple one; in the example given in Fig. 93 the silencer can be readily dismantled by unscrewing the nuts (A) shown, and detaching the central drilled tube.

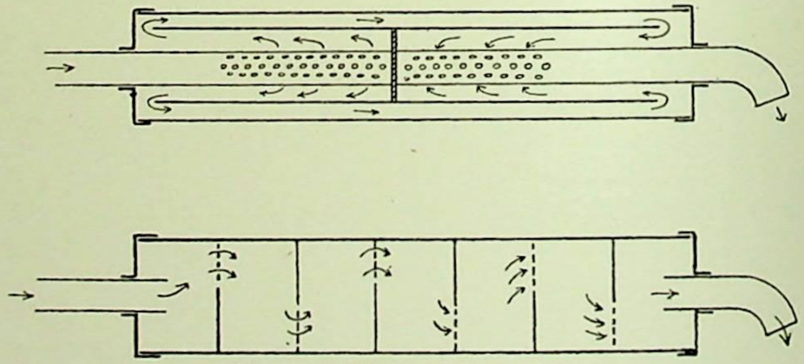


Fig. 94.—Types of Silencers.

It is very easy to silence effectively an engine by inserting enough obstruction to the gases, but at the cost of back pressure; this entails a serious loss of power, and frequently causes overheating. It is not possible to silence effectively an engine without a small loss of power; careful tests, however, have shown that this loss of power need not exceed from 1½ to 4 per cent. for very good silencing. Fig. 94

shows some alternative methods of silencing the gases employed in practice.

The ideal silencer would allow the exhaust gases to expand gradually as they cool, and to issue in a series of continuous streams, or in one stream only; for this reason the capacity of the silencer should be ample enough to accommodate the cooled gases, whilst the hottest gases should be led past suitable surfaces for radiating the heat, i.e. along the inner surface of the silencing box.

In aircraft practice the silencing pipe is now made very long and of large diameter, as shown in Fig. 95; it is closed at the end remote from the engine, and is provided with a large number of holes along the latter part of its length; it is also placed in the slip stream of the propeller for cooling purposes. This practice is worth noting.

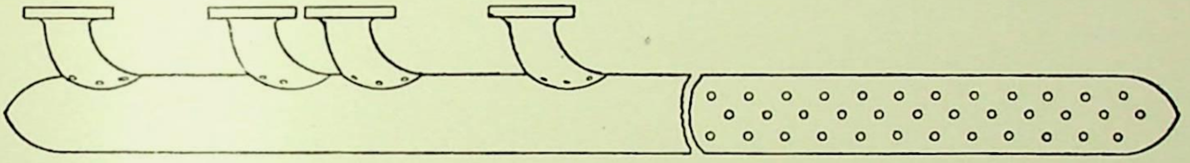


Fig. 95.

The capacity of the silencer should be such that for every brake horse-power there is not less than ·008 cubic feet (or 13·8 cubic inches) of silencer-chamber volume. The weight of the silencer should not exceed about 0·2 to 0·3 lb. per b.h.p.

THE STARTING HANDLE

Although the majority of modern automobile engines are now fitted with electric-motor starters, in place of the earlier handle-starting arrangements, it is usual to include a detachable handle for starting up or cranking over when the engine is cold, and for occasions when the engine is being adjusted. The starting handle arrangement of the Dorman engine can be clearly seen in Fig. 97, whilst an example of a hinged starting handle, as used upon certain commercial vehicles, is shown in Fig. 96. This arrangement enables the handle to be stowed away, after starting up.

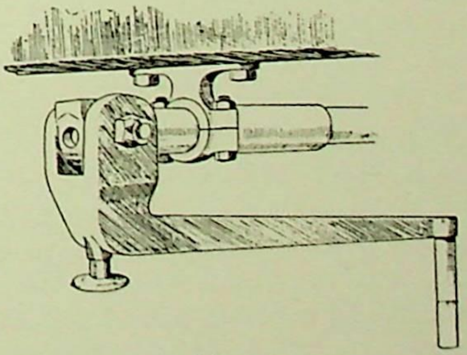


Fig. 96.

The forward end of the crankshaft is provided with a small extension, having a series of teeth, consisting of sloping faces alternating with straight or axial ones. The starting handle is provided with similar teeth, and is normally kept out of engagement by means of a spiral spring. On engagement the handle drives the crankshaft by means of the axial faces of the teeth, but when the engine fires, it overruns the starting handle and the sloping faces on the crankshaft portion force the starting handle away and out of engagement. The net result, then, is that the starting handle can only rotate the engine in one direction, and is disengaged when the engine fires. If, however, the engine *back fires*, that is to say, rotates for a short period in the wrong direction, the starting handle is carried around positively with it, and there is a serious risk of the operator's wrist being sprained or broken. Care should therefore be taken to hold the handle with the hand only partially closed, so that the handle comes away readily in the event of a back fire. There are one or two automatic devices on the market to obviate the risks attending a back fire.

THE COMPLETE ENGINE

The principal components of the modern petrol engine have now been described, but certain parts have intentionally been omitted, in so far as detailed descriptions are concerned. Amongst these may be mentioned the oil pump and lubrication system, magneto, sparking plugs, electric or mechanical starter, water pump, tyre pump, and similar accessories; these will be referred to in detail in subsequent chapters.

Fig. 97 shows a side sectional elevation of the Dorman 10 h.p. engine, and will serve to illustrate the disposition of the various components hitherto described, in the case of a well-designed modern four-cylinder engine.

Commencing from the extreme left, the starting handle, with its long bearings, helical spring, and two engaging teeth, will be clearly evident; the corresponding teeth on the crankshaft are also discernible, immediately in front of timing-shaft driving gear. The crankshaft shown is of the four-throw, three-bearing type, with two inclined and four straight crank-webs. At the rear end is an oil thrower, the purpose of which is to prevent the escape of the lubricating oil along the flywheel bearing and out of the crankcase. At its near extremity the crankshaft has a large flange to which the flywheel is bolted.

The camshaft is a steel shaft with the cams machined from the solid, and is of large diameter and therefore very stiff and sturdy. The cams are ground, after hardening, to a profile which combines a high lift with efficient valve opening and quietness in operation. Silent helical gears are used for the timing gears, and the larger diameter gear on the left side of the camshaft is provided with an extension to carry the belt pulley for the radiator cooling fan and the dynamo drives. A two-bladed fan, with its spindle running in ball bearings, is provided.

The connecting rods are made from H-section steel stampings, with die-cast white-metal bearings at their big ends; the usual caps and oil scoops are employed at the big ends.

Each gudgeon pin is secured in the small end of its rod by means of the clamping arrangement shown in the side elevation (Fig. 98), so that the gudgeon pin oscillates in bearings formed in the piston bosses, the latter having holes drilled through them to convey oil to the bearing surfaces. These oil holes can be seen in the section of the piston shown in the left-hand cylinder in Fig. 97. Each cast-iron piston has three cast-iron rings around the head and a groove at the bottom of the skirt which serves to collect any surplus oil on the cylinder walls and thus reduce carbonisation. The importance of keeping the piston weight down and of ensuring uniformity of weight has been carefully considered in this design, and each piston is separately weighed and passed, if satisfactory, before assembly.

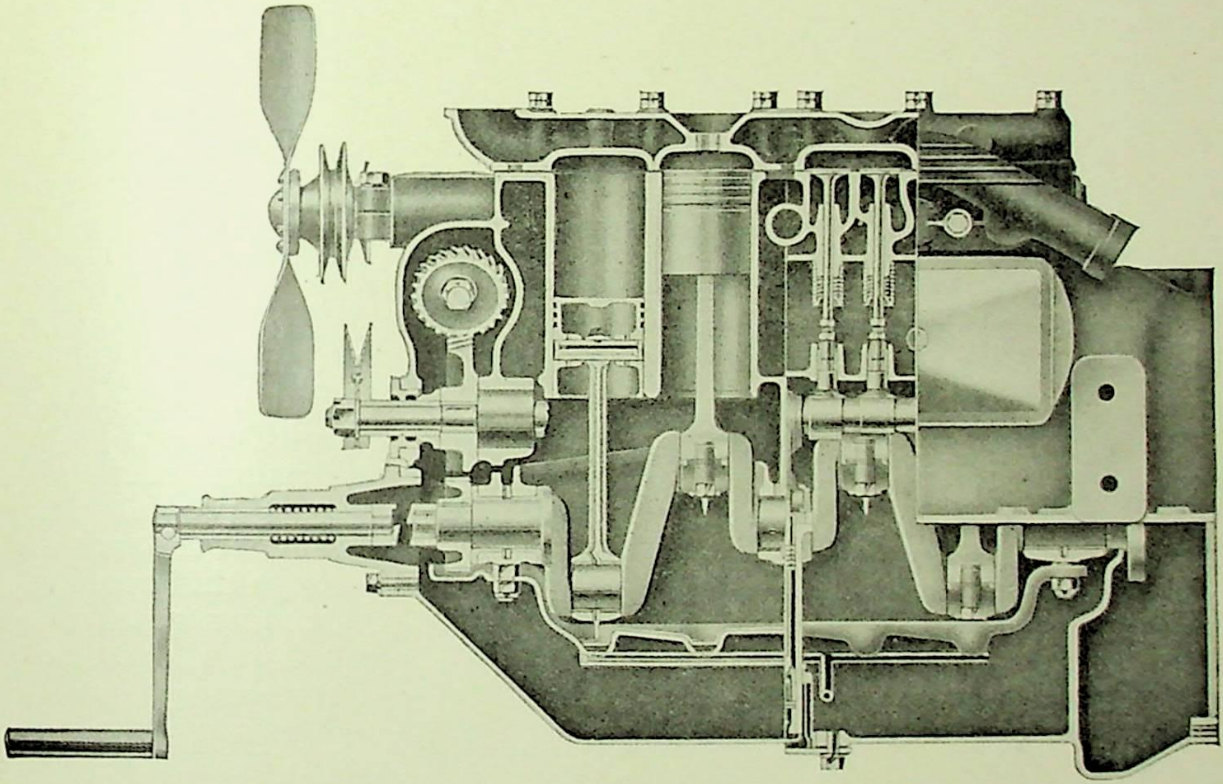


Fig. 97.—The Dorman 10 h.p. Engine (Side Sectional Elevation).

The front sectional illustration in Fig. 98 shows the valve gear. The valves are of the side-by-side type, and are operated directly from the camshaft, through well-proportioned tappets which engage with the cams. Each tappet is provided with an adjusting screw and lock nut at its upper end in order to allow the clearance between it and the valve stem being carefully adjusted. The valves, both inlet and exhaust, are interchangeable, and are enclosed on the left-hand side in a chamber having openings in its floor communicating with the crankcase. By this means a portion of the oil mist which exists in the crankcase finds its way up into the valve chamber and serves automatically to lubricate the valve stems and tappets.

The accessibility of the valve tappets and valves is a good feature of this engine, and is a point which will be appreciated by those who have tried to remove valves in small monobloc engines. The engine described is of monobloc cylinder construction, the cylinder unit having a separate detachable head, as will be seen from the illustrations given. The upper half of the crankcase is formed integrally with the cylinders, and is made of cast iron. The timing gear, flywheel casing, and the water jackets are also integral with this casting. There is no separate front cover for the casing of the timing gears, but instead, a large opening with a cover plate supporting the oil filler is provided at the side of the casing. Through this opening the timing gears are passed in the course of erection.

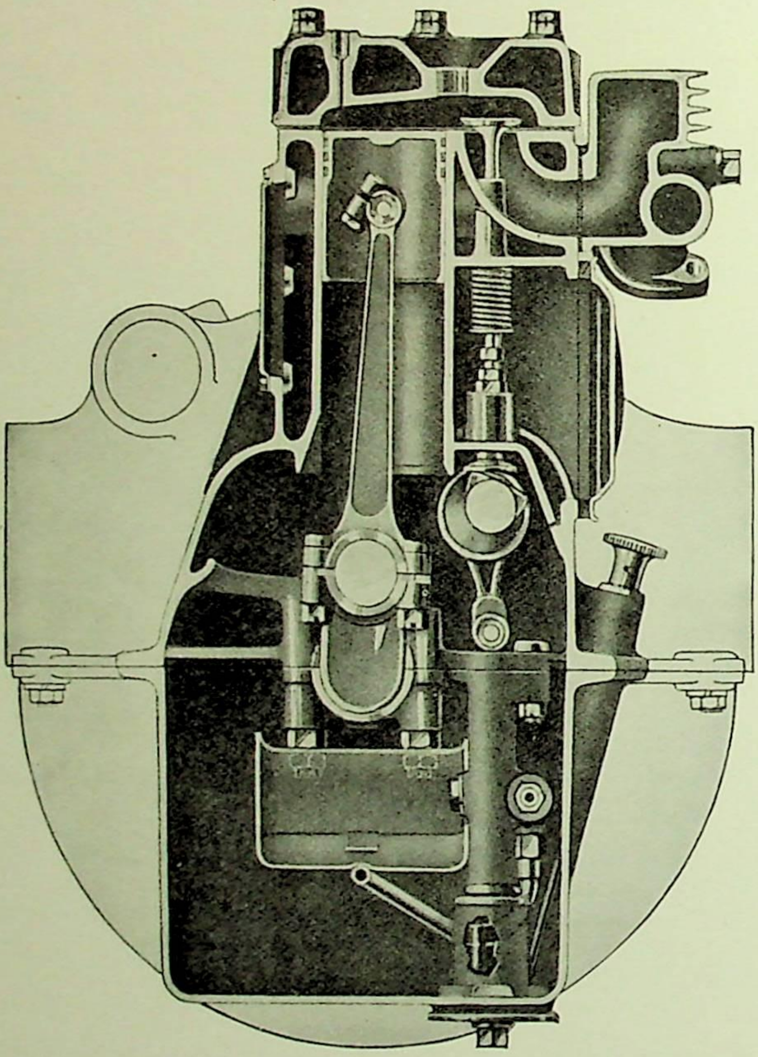


Fig. 98.—Front Sectional View of the Dorman 10 h.p. Engine.

The one-piece cylinder head, as we have stated, is a detachable casting, as also is the oil-sump portion; either can be readily removed without disturbing the other part, so that in the first case the combustion chambers and piston heads are accessible for decarbonising by the removal of a series of nuts which can be operated by an ordinary open-ended spanner, while in the other case the big-end and main crankshaft bearing can be easily reached by removing the oil sump.

The engine as a unit is suspended from the frame at three points, two at the rear and one at the front, the brackets at the back being formed at each side of the flywheel casing. The latter has a large circular flange to which can be bolted the front extension of the gear box, when this unit system of construction is employed.

The dimensions of the engine described are 69 millimetres bore by 100 millimetres stroke, giving a total capacity for the cylinders of 1,496 cubic centimetres ; the b.h.p. is 10 at 1,000 r.p.m., but as this engine is designed to run at speeds of 2,000 r.p.m. and above, the maximum horse-power is more than double the figure given for 1,000 r.p.m.

The subject of the lubrication arrangements of this particular type of engine will be discussed in the chapter dealing with the lubrication of petrol engines.

CHAPTER III

TYPES OF ENGINE

THERE is a variety of types and designs of petrol engine to be found on different motor vehicles; but in spite of this, the petrol engines in use may be divided into two broad classes—namely, (*a*) petrol engines of the poppet-valve type, but varying in the number and arrangement of the cylinders, and (*b*) petrol engines employing valve arrangements differing from the poppet-valve types, and varying also in the arrangement and in the number of cylinders.

It is proposed to deal with these two classes in the order named, and briefly to describe some of the more representative types of each.

The horse-power of a petrol engine is the governing factor in determining the dimensions and arrangement of the engine, but for a given horse-power it is usually possible to employ more than one arrangement of cylinders, and the deciding factor is then one of purpose. For example, it is possible to design an engine of nominal horse-power of say, 12, in a large single-cylinder unit, with a heavy flywheel and moving parts, or as a twin cylinder of smaller dimensions, or again as a four-cylinder unit with fairly small cylinders forming a light compact unit. It is possible also (and it has been done in one or two cases) to make a very light six-cylinder engine unit of this output. The single-cylinder unit would only be employed as a stationary engine, however, and would usually be fairly heavy. The twin-cylinder unit would be used on high-power motor cycles in the form of an air-cooled Vee twin, or horizontally opposed twin; it would be rather more costly to construct than a single-engine unit on account of the duplication of many of the machining operations, such as the connecting rods, pistons, valves, and cylinders. Again, the four-cylinder engine would be still more costly to manufacture under the same conditions; but this engine would be more compact, sweeter in running, more flexible, and could be made actually lighter on account of the more even torque and lighter reciprocating parts. This type would be employed on the more luxurious motor cycles and on light cars. In some cases manufacturers of light cars of the four-cylinder model have built a more luxurious car of about the same horse-power, but fitted with a six-cylinder engine instead of a four, so as to obtain more flexibility, acceleration, and smoothness of running.

In general, however, the horse-power output governs the type of engine, or at least the number of cylinders employed ; and the following may be taken as a rough guide in this connection :

Type of Engine.	Nominal h.p. at 1,000 r.p.m.	Cylinder Capacity (total) in cubic centimetres.
Single-cylinder two-stroke	2½ to 6	200 to 600
Single-cylinder four-stroke		
Small twin-cylinder four-stroke		
Twin-cylinder two- or four-stroke	5 to 10	500 to 1,000
Small four-cylinder four-stroke, air-cooled or water-cooled		
Large twin-cylinder opposed or Vee-type	10 to 15	1,000 to 1,500
Four-cylinder vertical		
Five-cylinder radial (air-cooled)		
Small six-cylinder unit (for light cars)		
Four-cylinder (normal car-engine)	15 to 25	1,500 to 2,500
Six-cylinder vertical		
Eight-cylinder Vee, or vertical (occasionally)		
Four-cylinder (occasionally and in American cars)	25 to 40	2,500 to 4,000
Six-cylinder (most commonly)		
Eight-cylinder Vee or vertical		
Twelve-cylinder Vee		

The larger number of cylinders for a given power usually denotes the engine of the more expensive car, or occasionally the speed or racing model of a manufacturer's car.

The present considerations will follow the lines of the above classification, commencing with the single-cylinder unit.

THE SINGLE-CYLINDER ENGINE

This type has already been referred to in the preceding chapter and several illustrations of it have been given. This model ranges from about 1½ to 6 h.p. (nominal) at 1,000 r.p.m., and is now invariably confined to motor-cycle engine types. The smallest engine which appears to be manufactured is of about 170 to 220 c.c., a typical example of the former having a bore and stroke of 60 and 60 mm., respectively, and is employed upon pedal-assisted motor-cycles, two-strokes, and four-stroke light-weight machines.

In the case of the four-stroke engine, either the side-by-side, the overhead, or the combination system of poppet valves is employed. The more powerful and efficient types employ the overhead-valve system, with either one or two exhaust valves per cylinder. An example of the latter type is shown in Fig. 10, which also indicates the push rods which work the valve rockers, and the adjustable heads on the push rods.

The arrangement of the flywheels has already been mentioned, but in passing it should be pointed out that in the case of one or two four-stroke engines and of the majority of two-strokes outside flywheels are employed. In these cases an ordinary solid or built-up single-throw crank of the one- or two-web type is employed.

The operation of the valves is very much the same as in the case of the car engines described. The crankshaft has a pinion on the side remote from the driving side, and this pinion usually gears into another having twice the number of teeth; this latter wheel frequently has both inlet and exhaust cams cut on an extension and side by side. Rockers pivoted on the timing case and crankcase engage on the one side with the cams and on the other with the valve tappets.

In certain instances two timing wheels and cams are employed, one for the inlet and the other for the exhaust, whilst in other cases a single cam operates both valves through rocker arms. The wear on the cam is of course increased in this instance. The cams can be of the external type described in the last chapter, or may be cut internally, that is to say, inside the rim of the timing wheels.

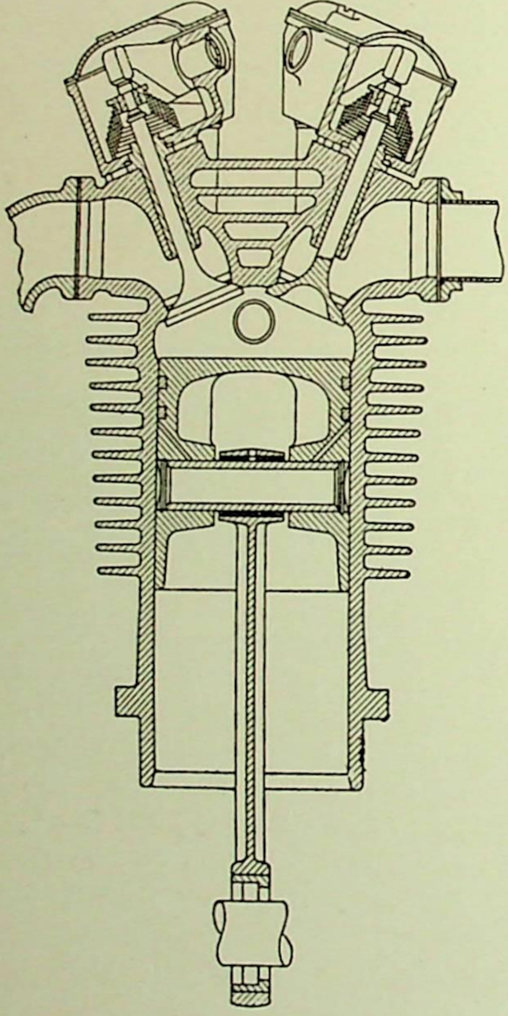


Fig. 99.—Overhead-valve Air-cooled Engine Cylinder (*I.A.E.*).

It is the tendency of modern motor-cycle engines to be fitted with detachable cylinder heads and aluminium pistons, and to have the valve gear encased so as to reduce the noise of the tappets or rocker arms.

A section through an overhead-valve engine cylinder of the air-cooled type is shown in Fig. 99, which shows also the piston and connecting rod. In this case, the valves are inclined, and are operated by overhead-valve rockers and push rods; a special form of casing totally encloses the valve gear and push rods, so that the latter are kept warm and expand with the cylinder to a certain extent. Volute-type valve springs are employed, as these enable the springs to be very short and compact, compared with the usual helical coil springs. In some cases leaf-springs, similar in design to the semi-elliptic and cantilever springs employed for the chassis springing, are employed for the valves in order to minimise space. In most cases return springs, quite separate from the main valve springs, are employed to return the tappets or push rods and gear during the valve-closing periods.

THE TWO-CYLINDER ENGINE

There are four practical methods of arranging the two-cylinder engine, namely: (1) two-cylinder vertical with cranks opposed; (2)

two-cylinder vertical with cranks in same line, or a common crank; (3) two-cylinder opposed, with cranks opposite; and (4) two-cylinder Vee-type with a common crank pin.

The first two methods, which are illustrated in Figs. 100 and 101 respectively, are very seldom employed, for the reason that the balance of the reciprocating parts is bad and cannot conveniently be improved.

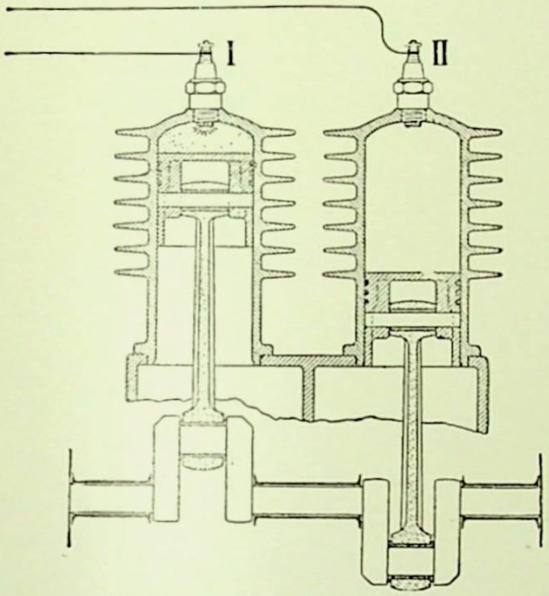


Fig. 100.—Cranks at 180°.

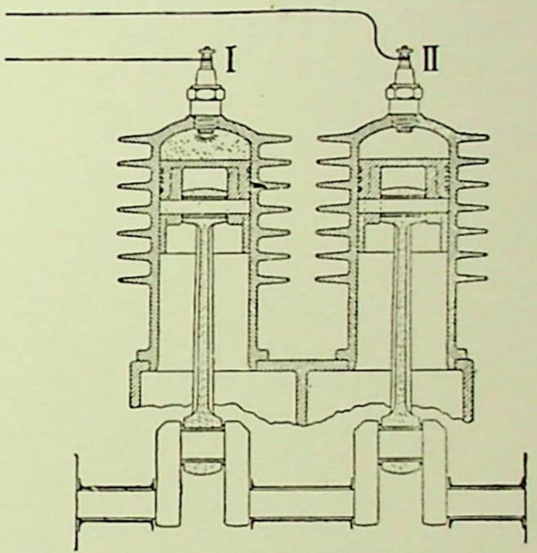


Fig. 101.—Cranks Coincident.

With the arrangement shown in Fig. 100 in which the pistons move up and down alternately, the balance of the reciprocating parts is better than in the other example, but a rocking couple is introduced which tends to rock the engine fore and aft. The power impulses also occur at unequal intervals. With the latter arrangement the balance is equivalent to that of a single-cylinder engine (which is bad), but the power impulses occur at regular intervals.

In the case of the two-cylinder opposed engine, not only are the reciprocating forces very nearly in perfect balance, but the explosions occur at regular intervals. This type of engine is now widely employed, on account of its excellent balance and even pulling properties; very high engine speeds are obtainable owing to the balance and uniformity of the impulses. Fig. 102 illustrates, in outline, the principle of the opposed engine and shows how the two pistons and their con-

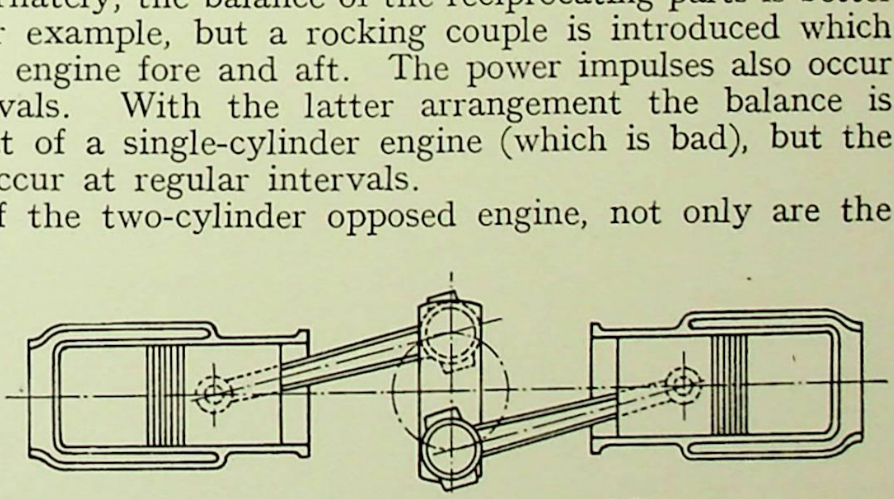


Fig. 102.—The Horizontally-opposed Engine.

necting rods move inwards and outwards together, in such a manner that their reciprocating forces are balanced when the axes of the opposed cylinders are in the same line. It is generally not possible, for reasons connected with the manufacture, to arrange for co-axial cylinders, so that there is a small rocking effect, introduced by the cylinders being a little out of line, which tends to rock the cylinders in a direction perpendicular to their axes, that is, sideways.

An excellent example of the horizontally opposed twin-cylinder engine is that of the 8 h.p. Rover unit, the engine being illustrated in section, on the left-hand side, and in outside elevation, on the right-hand side, in Fig. 103. This air-cooled engine is of 85 mm. bore and 88 mm. stroke and is of 10 h.p. nominally, the cylinders being of cast iron with detachable heads. The cylinder heads are held down by three short studs each, instead of long studs from the crankcase and bridge pieces, as it is found in practice that the latter tend to crack the cylinder head owing to the differential expansion effect which occurs.

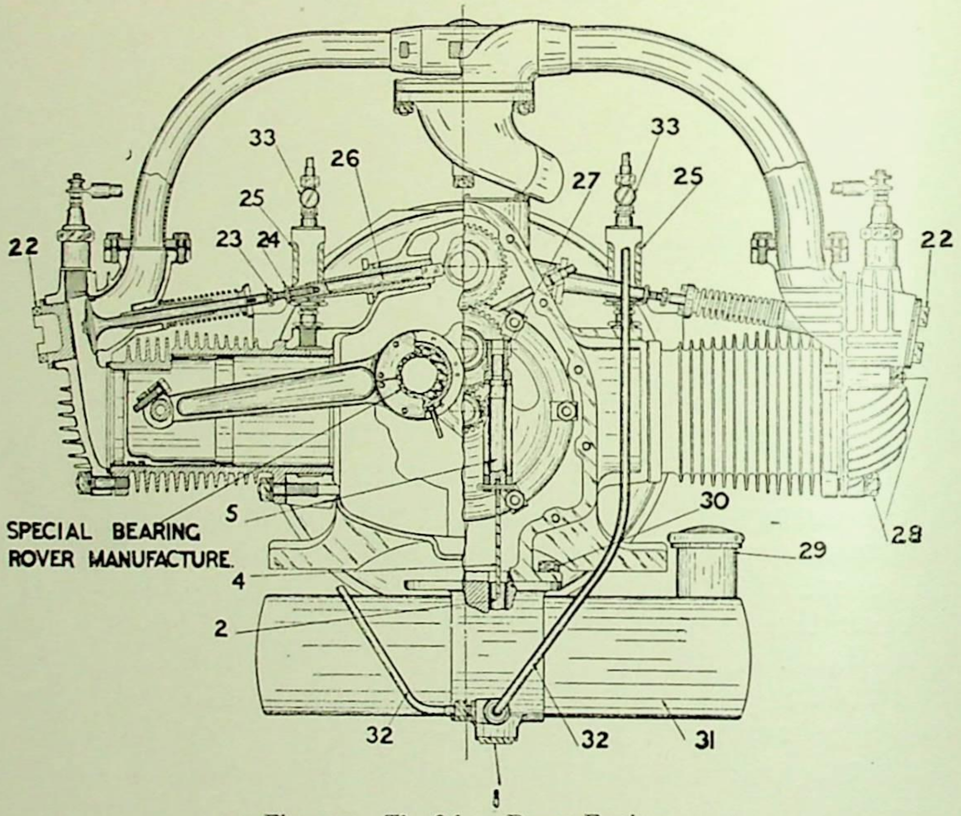


Fig. 103.—The 8 h.p. Rover Engine.

The valves are of the side-by-side direct type, the exhaust valve at the front end so as to receive the full benefit of the cooling draught. The valves are also inclined with the heads downwards, as shown in Fig. 103. Unusually long tappets, with rounded ends, are employed, which bear direct on the cams, a part of the inlet cam being visible in the left-hand view of Fig. 103. The camshaft is mounted above the crankshaft, and runs in ball bearings, each bearing taking the thrust in one direction. The cylinders are provided with extra long registering portions which fit in corresponding parts of the crankcase and are held thereto by four studs and nuts, one of which is shown in the illustration on the lower side.

The pistons are of cast iron, of the flat-topped, two-ring type, with the gudgeon pin clamped to the small end of the connecting rod, and arranged to rotate, or rather to rock, in the piston bosses. Brass buttons are fitted, one at either end of the gudgeon pin, to prevent cylinder scoring, should the gudgeon pin work loose. The design of the connecting rods is of some interest, in view of the previously mentioned point relating to the importance of obtaining coincidence of the cylinder axes. In this case, the offset of the cylinders is reduced as much as possible by employing roller bearings in the big ends of the connecting rod, so that their width, perpendicular to the cylinder axes, is as small as possible. The roller bearings themselves, which are of the twin-roller type, are offset in relation to the pistons, and they run directly upon the case-hardened and ground crankpins. The two rows of rollers are separated by a thin washer and run in a hardened race let into the big end. They are prevented from moving laterally by two special washers fixed by means of rivets passing through the connecting rod.

The main bearings for the two-throw crankshaft consist of twin-journal ball bearings at the rear or driving end, where the flywheel is, which are arranged so as to take the thrust of the clutch spring, whilst the front bearing is of the single-journal type.

It will be seen from the illustration shown in Fig. 103 that the sparking plug is directly over the inlet valve, and that the carburettor is situated immediately above the axis of the crankshaft, the mixture being led into the cylinders along the curved induction pipe shown.

A general arrangement, in section, of the crankshaft, camshaft, and single-plate type clutch is shown in Fig. 104. This illustration shows also the journal and roller bearings, and the camshaft (with its four separate solid cams) immediately above the crankshaft, together with the single-row ball races at its ends. An extension of this camshaft drives the magneto, which provides a spark once every revolution of the engine.

The starting handle is provided with substantial bearings widely separated ; the engagement teeth can be seen in the diagram, being shown by the crosses.

The flywheel is keyed on to the tapered rear end of the crankshaft, a Woodruff type of key being used. The single dry-plate type clutch shown consists of three parts, namely the Raybestos-faced flywheel, the thin sheet-steel driven disc, which is rigidly attached to the gear-box shaft, and a floating Raybestos-lined member behind the disc. The size of the Raybestos friction material used is about 1 inch in width and

8¾ inches diameter; the necessary normal pressure between the floating disc and flywheel friction material and the driven disc is obtained from four springs, two of which are shown near the periphery of the flywheel.

Although the subject of lubricating systems is dealt with more fully later, yet in this case it will be better to describe the present system here, on account of the convenience of the diagrams, and of the particular nature of the method employed. In this example oil is pumped from an oil tank under the crankcase to two drip-feeds, one over each cylinder as shown in Fig. 103, and also to the timing case. The oil thus lubricates the cylinder walls and the pistons direct, and the surplus oil flows from the walls and timing case into the bottom of the crankcase and there forms an oil bath into which the dippers of the big ends pass and obtain oil for lubricating the roller bearings. When the oil bath (shown in Fig. 104) is filled to a certain predetermined level, any excess of oil flows down an *overflow* pipe through a gauze strainer, which collects any solid matter, and thence into the oil tank again, where it is cooled by the air passing over the outside and is ready for further use. The pump employed is of the sliding-vane rotary type driven by means of a worm formed on the left-hand side of the intermediate timing-gear wheel.

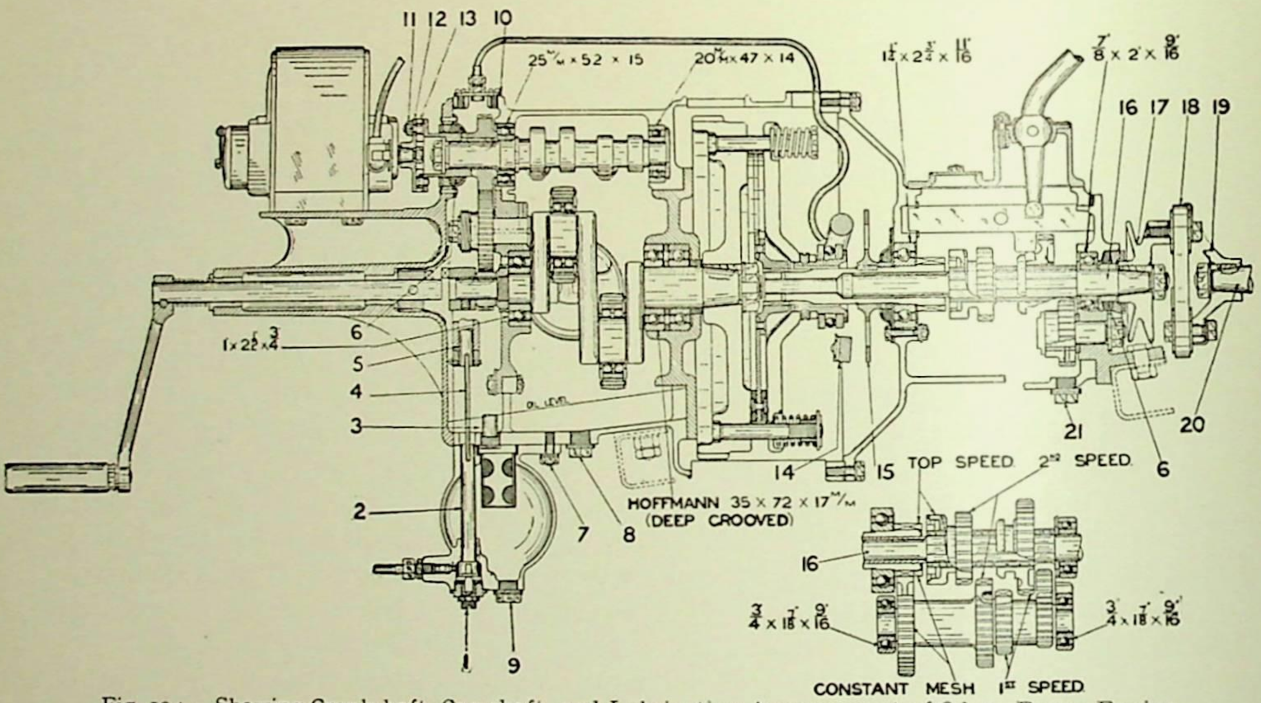


Fig. 104.—Showing Crankshaft, Camshaft, and Lubrication Arrangement of 8 h.p. Rover Engine.

THE VEE-TYPE TWO-CYLINDER ENGINE

As this same principle is employed in connection with eight- and twelve-cylinder Vee-type engines, it will be briefly discussed here. Fig. 105

shows diagrammatically the general arrangement of this type of engine. It will be seen that the two cylinders are inclined at an angle to one another. This angle varies in practice from 40° to 90°, the latter giving the better balance and firing impulses.

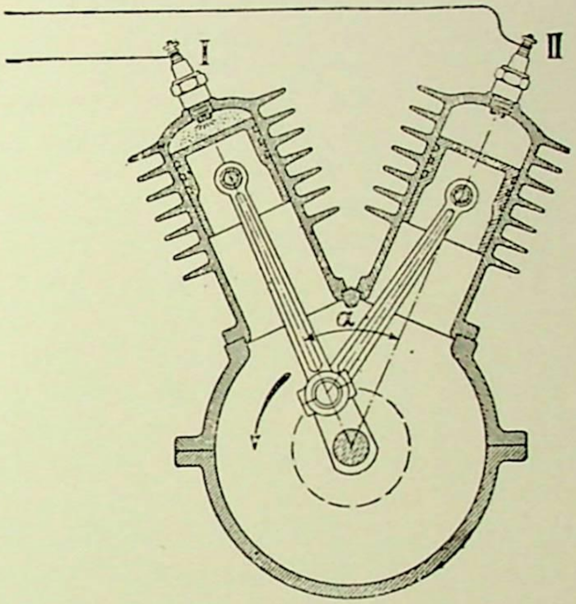


Fig. 105.—Vee-cylinder Engine.

The balance of this type of engine is not nearly so good as that of the opposed type, and it is only possible to make a kind of compromise by partially balancing the reciprocating forces by means of a balance weight on the opposite side to the crankpin. Usually, the flywheels are cast with a heavier portion opposite to the crankpin, for balance purposes.

The balance of the 90° twin-cylinder engine is better than that of most other engines of this type, the out-of-balance force being less than with smaller angles between the cylinders. Thus in the case of a 60° twin engine the unbalanced force is nearly three times that of the 90° twin, and the amount (or amplitude) of the vibration caused is approximately twelve times that of the 90° twin engine. The firing intervals in the case of the 60° twin are 300° and 420° and for the 90° twin 270° and 450°, so that the former type is rather better in this respect. However, the advantages in the matter of the balance of the 90° twin considerably outweigh its less regular firing periods, and in general it is superior to the engines with smaller included angles between the cylinders.

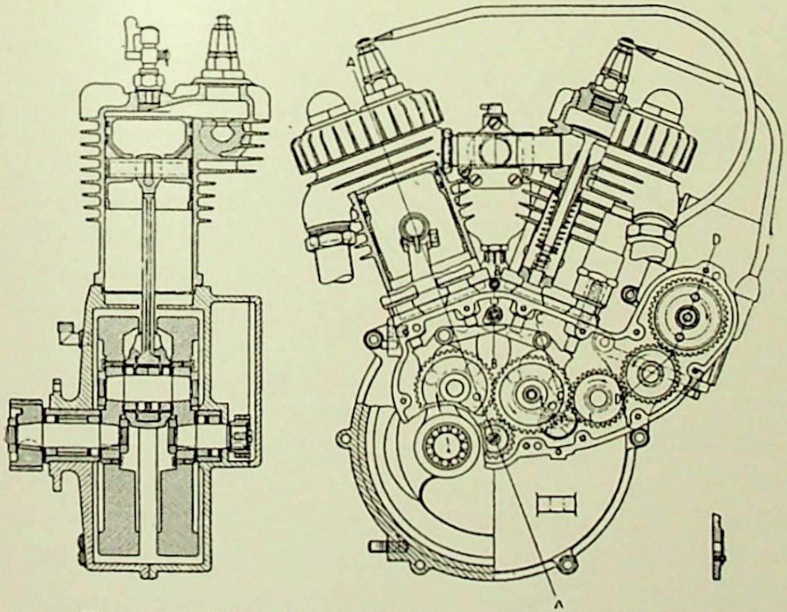


Fig. 106.—The 4 h.p. Indian Vee-twin Engine (*A.E.*).

Fig. 106[1] gives a side and front sectional elevation of the Indian twin motor-cycle engine of about 4 h.p. (nominal). The bore and stroke are 70 mm. and 78 mm. respectively, the cylinder capacity being 596 c.c. The maximum horse-power obtained by tests from this engine is about 11 b.h.p.

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[1] By courtesy of *The Automobile Engineer.*

Referring to the right-hand illustration, the cylinders are set at an angle of 42°, which is a fairly small angle for this class of twin. The cylinders are of cast iron, and are not of the detachable-head type.

The cooling fins are of the usual type around the barrel, and are radial with the axis of the cylinder on the cylinder head. The shape of the combustion chamber is rather less flat than in the usual design of L-headed engine, and it is very probable that better turbulence and combustion efficiency are the result.

The cylinders are provided with barrel-type registers in the crank-case, and are held down by means of four short studs. The valves are of the usual type, and the valve springs are enclosed in telescopic cylindrical cases, to keep out the dust and reduce the noise of the tappets.

The arrangement of the timing gears will be evident from Fig. 106 (right-hand view). The small gear wheel on the crankshaft meshes with two wheels of twice the number of teeth, which are situated above it. Each of these latter wheels has a cam formed solid with it, and the one cam operates the two valves through the rocker arms shown, which are provided with rollers to run on the cams. One rocker arm lies on the right and the other on the left-hand side of the cam-wheel's centre line, the pivot (or centre bearing) for each rocker being just above the cam wheel. The rocker arm acts directly upon the tappet end.

The piston is of cast iron, with two rings above the gudgeon pin, and there is a groove machined in the bottom of the skirt to collect the oil. The lower piston ring acts as an oil scraper, and small drilled holes in the lower ring slot conduct oil to the gudgeon pin, which is of the fixed small-end type, so that the pin floats in the piston bosses. The clamping bolt can be seen in the left-hand cylinder in Fig. 106. Any possible end-movement of the gudgeon pin is prevented by the clamping bolt, which, as will be seen in the left-hand diagram, engages in a circular groove turned in the centre of the gudgeon pin. The big ends of the H-section connecting rods are fitted with roller bearings, as are the two main crankshaft bearings, and the sides of the flywheels next to the roller bearings are provided with hardened steel facings.

In connection with the timing-gear wheels, the right-hand wheel drives the magneto through the intermediary of two idle pinions.

The exhaust valves in most motor-cycle engines are provided with an additional and separate lifting gear controlled from the handle bars, either by means of Bowden cable or by a series of jointed rods. The object of this exhaust-valve lifter is for the control of the engine by the rider, for by lifting the exhaust valve the engine can be quickly put out of action. In the present example, there is a single cam worked by a lever, which can depress both exhaust rocker arms, extensions being provided for this purpose. The exhaust lifting cam can be seen in the right-hand diagram of Fig. 106 immediately above the central timing pinion and just over the extensions of the exhaust rockers between the exhaust-valve tappets.

The engine is positively lubricated (as distinct from the usual splash system fed by a drip feed), the oil pump being driven from one of the timing-gear pinions through a small worm and worm-wheel. The pump is of the reciprocating plunger type. Oil is pumped from the tank into the crank chamber by way of a hole drilled in the wall of the timing-gear chamber, non-return valves being fitted to the pump inlet and outlet. The delivery pipe from the pump is connected to a right-angle piece, which is screwed into the crankcase wall.

The construction of the flywheels and the built-up crankshaft will be followed from the left-hand sectional view. For assembly purposes, the crankcase is made in two halves which are bolted together by seven bolts; one half of the crankcase has the timing-gear casing and the magneto bracket cast integrally with it.

The foregoing description is one of a typical design of Vee-twin, and covers the general methods adopted for motor-cycle engines of this type.

THE FOUR-CYLINDER ENGINE

This is by far the most widely employed type of motor-car engine, and has steadily increased in popularity during recent years. Many manufacturers of cycle cars and light cars who previously fitted twin engines are now favouring the four-cylinder type, whilst several six-cylinder car manufacturers are discarding their engines in favour of the four.

The four-cylinder engine represents an excellent compromise in the matter of even torque, very good engine balance, and in construction.

The general arrangement of a four-cylinder engine of the two-cylinder block type is illustrated diagrammatically in Fig. 107, the cylinders being numbered 1, 2, 3, and 4. It will be seen that the two extreme pistons move together, whilst the two inner ones also work up and down together, but exactly opposite (i.e. 180°) to the other two.

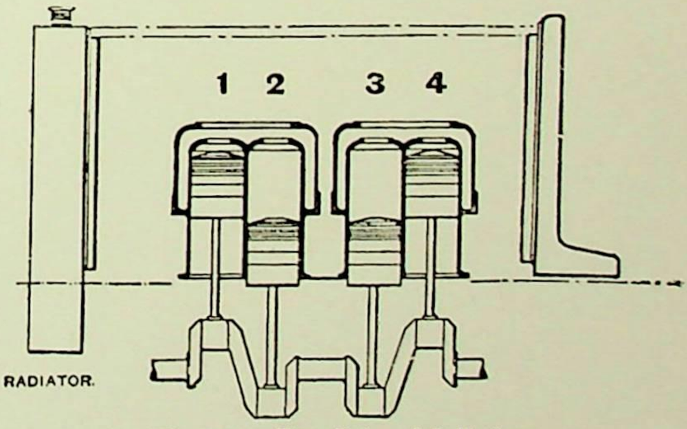


Fig. 107.—Two Pairs of Cylinders.

In this manner, if the pistons and connecting rods are all of equal weight, it will be evident that in all positions the two outer ones practically balance the two inner ones, so that the reciprocating forces are always in approximate balance in the vertical sense. They are not, however, completely in balance, owing to the fact that the angularity or obliquity of the connecting rods does not give an exact symmetrical arrangement of the pistons, as will be seen from Fig. 108, which represents the corresponding piston positions for the equal crank-positions shown.

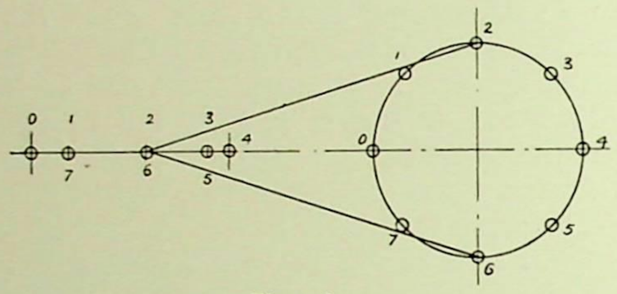


Fig. 108.

If the connecting rods were very long in comparison with the crank arms, the symmetry would be almost perfect and the four reciprocating forces would always be in balance.

It can be shown mathematically that the out-of-balance effect in a four-cylinder engine of the type described is equivalent to a mass of weight equal to $\frac{M.r^2}{l}$, where M is the mass of the reciprocating parts of one cylinder, r the crank radius, and l the length of the connecting rod.

This mass reciprocates up and down in the direction of the cylinder axes, at *twice* the engine speed, so that it gives rise to a vibration of frequency equal to twice the number of revolutions per minute. This unbalanced secondary force, as it is termed, depends for its magnitude upon the weight of the complete piston and the smaller end of the connecting rod, and also upon the ratio of the crank to connecting rod. This unbalanced force may be reduced in magnitude by making the reciprocating parts, such as the piston and gudgeon pins, as small as possible and by employing longer connecting rods. It is in the former direction that most benefit has been obtained, namely, in the use of aluminium-alloy pistons, and special alloy-steel connecting rods and gudgeon pins.

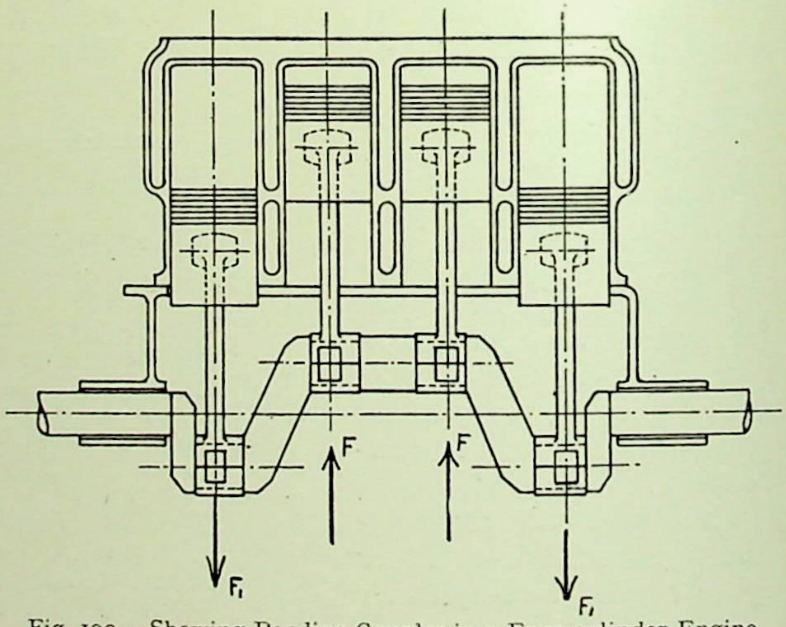
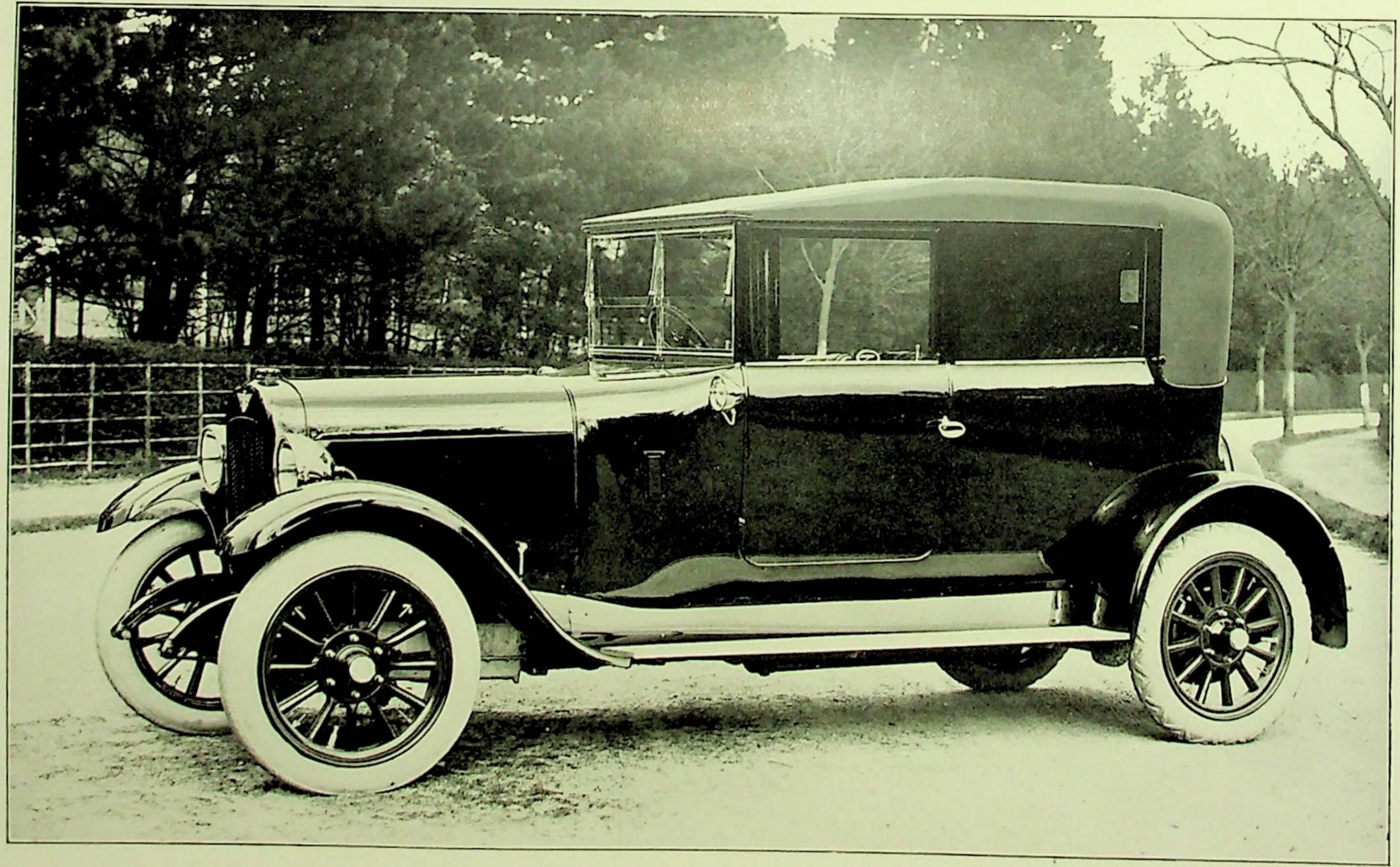


Fig. 109.—Showing Bending Couples in a Four-cylinder Engine.

PLATE IV



THE 20 H.P. AUSTIN ALL-WEATHER CAR.

In the case of a four-cylinder engine of 4-inch bore by 5-inch stroke, with connecting rods of length 10 inches, the complete pistons in cast iron will weigh about 3½ lb. each and the small ends of the connecting rods about 1 lb. each, making a total of 4½ lb. each for the reciprocating parts. The greatest value of the reciprocating force for any one piston, at 2,000 r.p.m., works out at about 1,285 lb., or about half a ton. The four reciprocating forces, however, balance one another, but the secondary unbalanced force works out at about 160 lb., and gives rise to a vibration occurring 4,000 times a minute.

It is possible to balance this vibrational force, and in one or two of the more expensive car engines, such as the Daimler, Vauxhall, and Lanchester ones, the method devised by Mr. F. W. Lanchester is employed with very satisfactory results.

Remembering that the unbalanced force oscillates up and down at twice engine-speed frequency, the principle of the Lanchester device can be readily followed. It introduces another reciprocating force of equal amount but in opposite phase, and of the same frequency as the unbalanced force.

The method of obtaining this reciprocating force is to employ two gear-wheels, meshing with one another, so that each rotates in the opposite direction. Each wheel is purposely set out of balance by an amount equal to one-half the unbalanced force to be neutralised. The net result of rotating two such weights W in opposite directions, as shown in Fig. 110, is to give a resultant up-and-down reciprocating force W, equal to the effect of twice the weight of either.

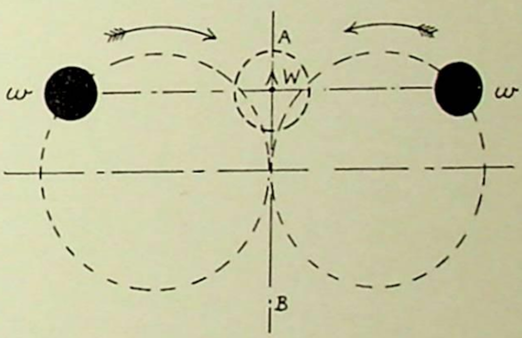


Fig. 110.—Showing how Two Rotating Weights can balance a Reciprocating Force.

The method of applying this principle to the Vauxhall 23–60 h.p. engine is shown in Fig. 111.

The helical gear wheel A, is fixed to a web of the crankshaft J, and engages the helical gear B, which meshes with a similar wheel C, so that both revolve in opposite directions, carrying with them their respective drums. These drums are mounted on a bracket E fixed to the crankcase and are fed from the engine's oil-pressure system by means of the pipes G shown. The drums are weighted in one-half, the exact amount for each engine being determined by a chart, the data for which is computed from a knowledge of the weights of the pistons, rings, gudgeon pins, and connecting rod. The gearing shown is arranged to rotate the weights B and C at twice engine speed, and, contrary to what one would anticipate at first thought, there is no power absorbed (other than by friction) in driving these weights, but the result is a very smooth-running engine free from the usual secondary vibrations.

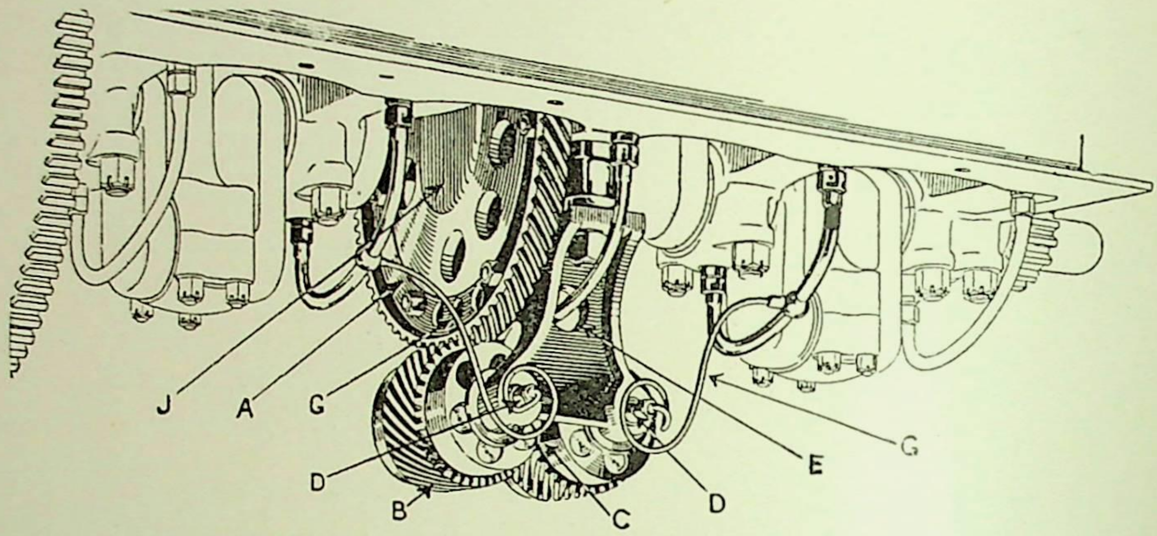


Fig. 111.—The Harmonic Balancer on the 23–60 h.p. Vauxhall Engine.

FIRING ORDER

The firing impulses, in the case of a four-cylinder engine, are uniform —that is to say, there is an equal interval between the firing strokes. These occur once every stroke, or twice per revolution, so that whenever there are two pistons at the top of their stroke, one is compressing the mixture prior to firing it and the other is exhausting the burnt gases.

The ignition apparatus, which is usually a magneto or a battery and high-voltage coil for producing the spark at the electrodes of the sparking plug, is arranged so as to provide one spark every stroke, that is to say, two per revolution, and at the appropriate moment when the pistons are very nearly at the end of their compression strokes. Using the same order of numbering of the cylinders as that shown in Fig. 107, the order in which the cylinders fire is as follows, namely, 1, 3, 4, 2. Although this is by far the most common order, yet there is another alternative method in which the cylinders fire in the order 1, 2, 4, 3.

A sectional view of a modern four-cylinder automobile engine of the side-by-side valve, detachable-head type has already been given in Fig. 97, the cylinders and heads being of cast iron. Fig. 112 is a front sectional view of the Dorman 12 h.p. engine of the overhead-valve type, with aluminium cylinder monobloc provided with a detachable head. In this case, the cylinder liners are of cast iron, and are machined all over. The pistons work on the inside surface, whilst the cooling water is in contact on the outside of the liner, and in the inside of the aluminium casting.

The sectional view shows clearly the line of detachment of the head just level with the top of the piston and the joint at the lower part of the cylinder casting, with the crankcase base forming the oil sump.

The dimensions are 69 mm. bore and 120 mm. stroke, giving a cylinder capacity of 1,790 c.c.; at 1,000 r.p.m. the engine gives about 13 h.p., and more than double this at the maximum speed.

This engine is of the type usually fitted to light cars, as it gives an excellent power-to-weight ratio, on account of its high efficiency and light construction.

The overhead valves are upon opposite sides of the cylinders and are operated from two camshafts, one upon either side of the engine's centre line. Adjustable push rods are provided. The ribbed exhaust manifold, for cooling the exhaust gases during their ejection, can be seen on the left-hand side. Another view of the overhead valves, taken with the cover plate removed, is shown in Fig. 113; it will be seen that the valves are staggered in position, and also that they are readily accessible. This view shows also the ribbed exhaust manifold held to cylinder by means of the two studs and nuts.

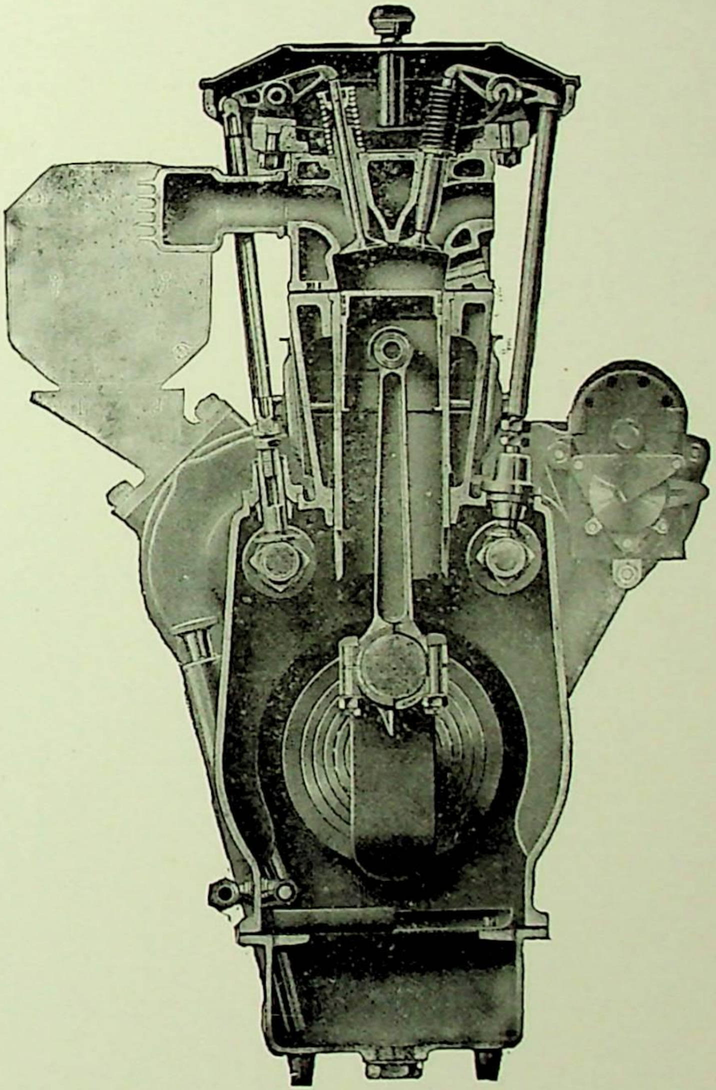


Fig. 112.—The Dorman 12 h.p. Engine.

Before passing on, mention should be made of the valve system in the Dorman engine, which is provided with an automatic lubricating system for the valve gear in the head of the cylinder block. The mechanism is enclosed by an overall cover secured in position by thumb nuts. From this chamber is formed a passage of large internal diameter which leads to the crankcase and constitutes a breather pipe; as a result there is a constantly rising oil mist passing from the crankcase to the overhead-valve mechanism, thus ensuring that the valve stems, rocking levers, and other parts are adequately lubricated. The balanced crankshaft of this engine is that illustrated in Fig. 85.

Having dealt with one or two typical four-cylinder engines, in section, it may be of interest to refer to some of the external features of these engines, choosing for this purpose the 23–60 h.p. Vauxhall engine illustrated in Figs. 114 and 115.

This engine is of the four-cylinder vertical type, the bore and stroke being 95 and 140 mm. respectively.

The power obtained from this engine, which has overhead valves, together with that obtained from a side-by-side valve engine of similar dimensions, is shown by the following experimental results :

HORSE-POWER (BRAKE)

Engine Revolutions per minute	500	1,000	1,500	2,000
Overhead-valve Type	14·0	32·0	47·5	60·0
Side-valve Type	7·0	26·5	41·5	50·0

This overhead-valve type is, therefore, appreciably superior in the matter of power.

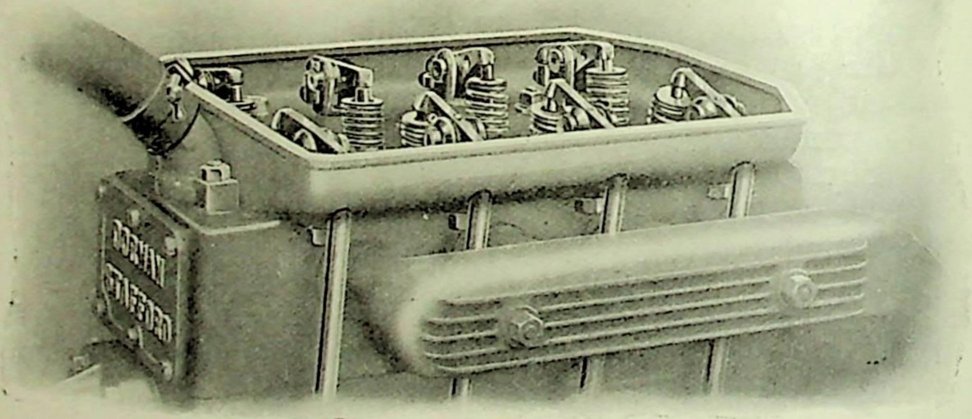


Fig. 113.—Showing Overhead Valves of Dorman 12 h.p. Engine.

Fig. 114 is an outside view of the engine from the near side, and shows the engine with the valve covers removed, and also the inspection doors in the crankcase, in front of the camshaft. The magneto and carburettor can be seen in this view. The former is held down, in correct register on the magneto platform cast with the aluminium crankcase, by means of the spring strap-fitting shown, for ease of removal purposes. The leather link-belt drive to the two-bladed cast-aluminium fan for drawing air through the radiator can be seen at the extreme left-hand side of the illustration in Fig. 114. The cylinder head in this type is detachable, and the water connection between the lower casting and the head is through large external aluminium ports, so that the joint between the two castings is a simple compression one, made by a plain aluminium plate.

The flywheel shown on the right-hand side in Fig. 114 is provided with teeth for the purpose of the starting motor shown on the left-hand side of Fig. 115. In this latter view, the engaging Bendix pinion of the starting motor is shown out of engagement.

The carburettor mixing chamber immediately above the carburettor is provided with a cast-aluminium hot-water jacket, which is supplied from the radiator water system. The object of this heated chamber is to assist in vaporising the mixture before it enters the inlet valve ports. The offside view of this engine shown in Fig. 115 gives a good idea of its appearance with the valve-gear cover in position; it shows the neat, clean design of a modern engine, in which nearly all of the mechanism is enclosed and unobtrusive.

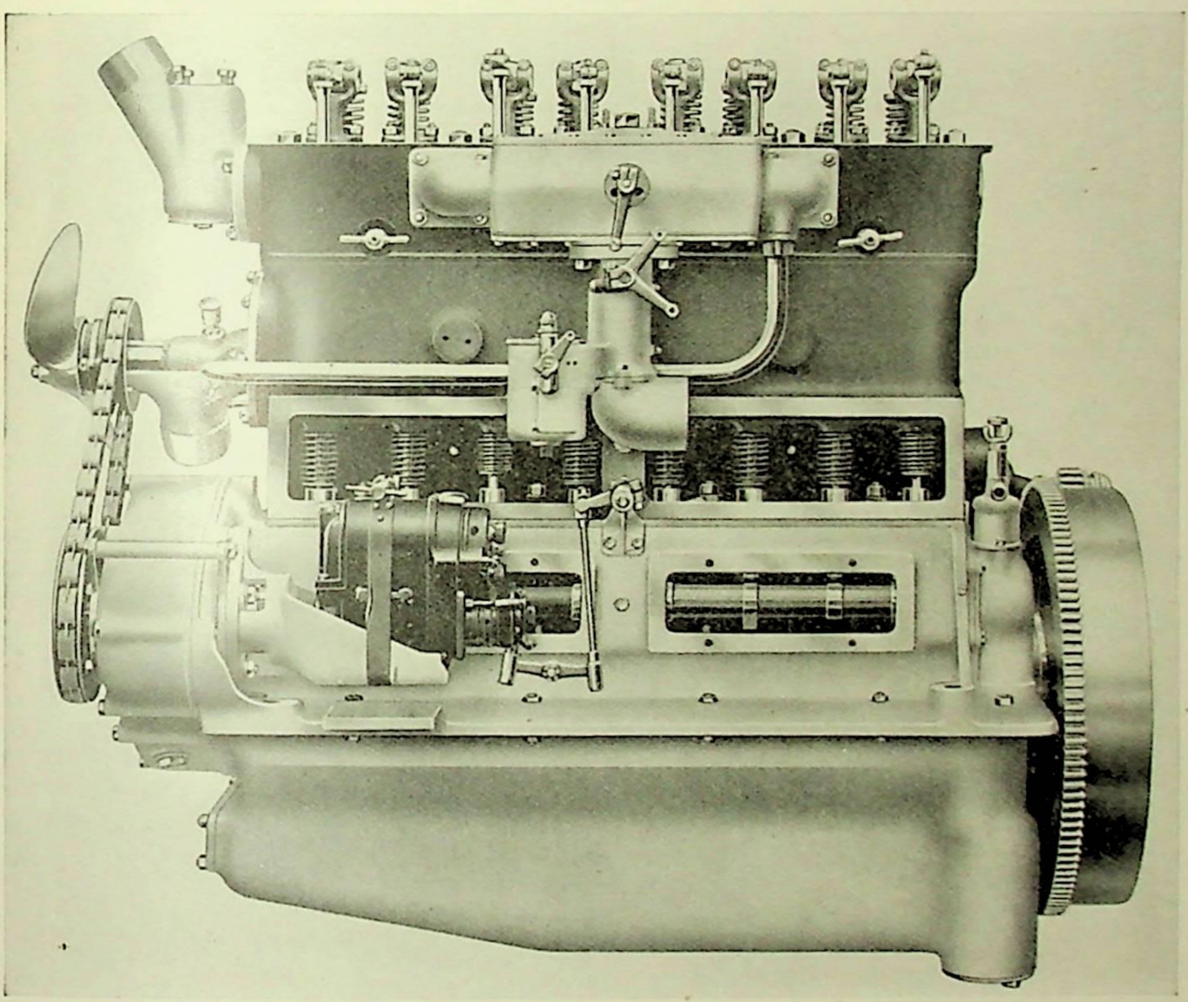


Fig. 114.—The 23–60 h.p. Vauxhall Engine (Carburettor Side).

The external cooling-water connections between the upper and lower cylinder castings, and the two-bladed aluminium fan are also apparent. The water outlet to the top of the radiator is shown above the fan. The outlet water is automatically controlled by a thermostatic device, fitted in the cylindrical portion, so that when the engine is cold the radiator is cut off, and the water in the jackets warms up more quickly. This device is described in the chapter on Carburation.

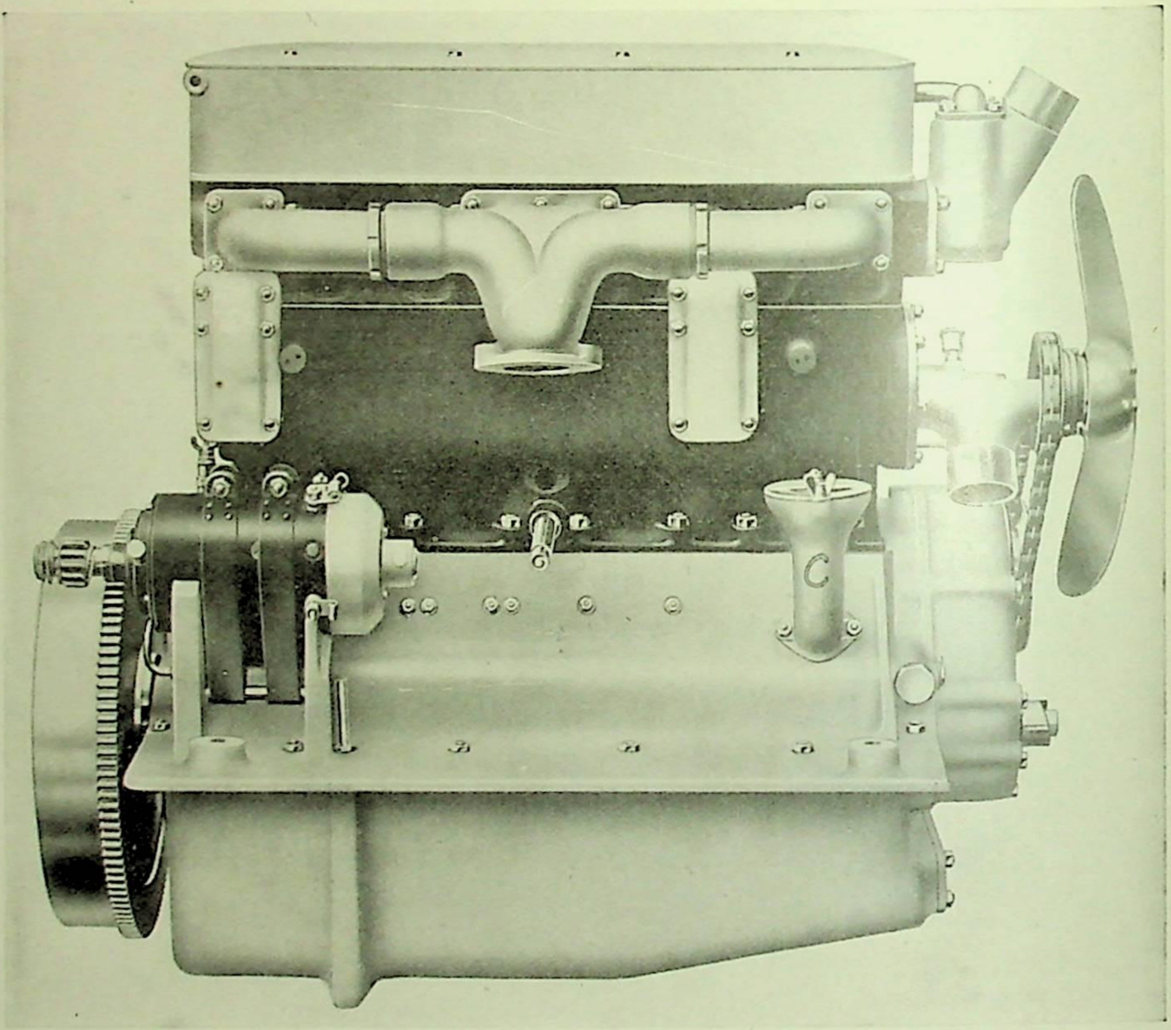


Fig. 115.—The 23–60 h.p. Vauxhall Engine (Exhaust Side).

The circulating water pump is situated immediately behind the fan, and the lower outlet is just below the fan connection to the engine. The engine is held to the chassis frame by means of the horizontal webs shown in the two illustrations, the holes for the holding-down bolts being shown at the ends of these flanges in the illustrations. One hole can be seen in Fig. 115 just below the oil-filler, C, on the right, whilst the other is beneath the starting motor on the left.

The engine illustrated is fitted with the Lanchester harmonic balancer, previously described.

HEAVY VEHICLE ENGINES

The four-cylinder engines hitherto described have been of the motor-car and light-car types. A brief reference will now be made to the heavy-engine type used on commercial vehicles.

As these engines are similar in principle to those already described, and differ chiefly in design for heavy work, and in their controls, it will not be necessary to devote much space to them here.

Fig. 116 shows a sectional illustration of the Dorman 40 h.p. engine of 120 mm. bore and 140 mm. stroke, giving a cylinder capacity of 6,326 c.c. This engine has an output of 40 b.h.p. at 1,000 r.p.m., when using petrol as a fuel, and 32 b.h.p. when using paraffin. It is usually fitted to motor vehicles, such as commercial lorries, chars-a-bancs, and similar vehicles, of from 3-ton to 5-ton load-carrying capacity.

The cylinders are of cast iron, bolted in pairs to the crankcase, which is divided horizontally on the crankshaft centre line; but the crankshaft bearings are supported solely by the upper half, the lower portion being detachable without disturbing the bearings, and forming merely an oil sump in which three gallons of lubricant are carried. The cylinders are machined by an initial rough-boring process, and are then ground in a special grinding machine which leaves them with a finely polished and accurate finish. The limits to which the cylinders and pistons are finished are narrow and such as to give good compression results, oil economy, and interchangeability.

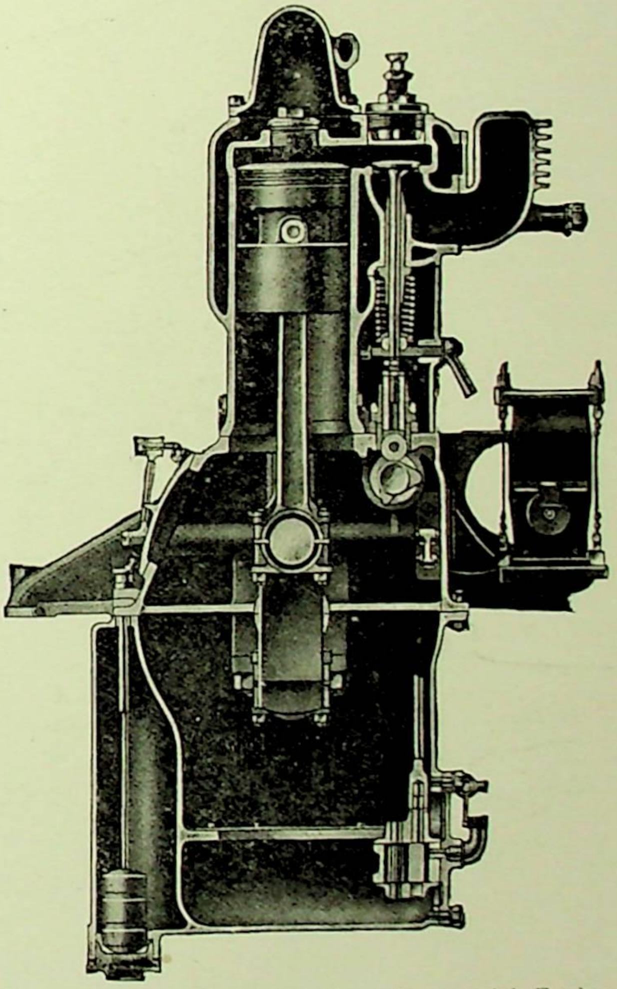


Fig. 116.—The Dorman 40 h.p. Commercial Engine (Front Section).

The cylinders of this type are usually cooled by the thermo-syphon process, but provision is also made for fitting a water pump. To each cylinder pair is bolted the half of a large-capacity water riser, which is made in either of two different angles as to the final outlet pipe, so that a direct flow can occur into the radiator inlet of the chassis into which the individual engine is fitted. The removal of these riser sections—which are coupled at the centre by a short length of rubber hose—allows

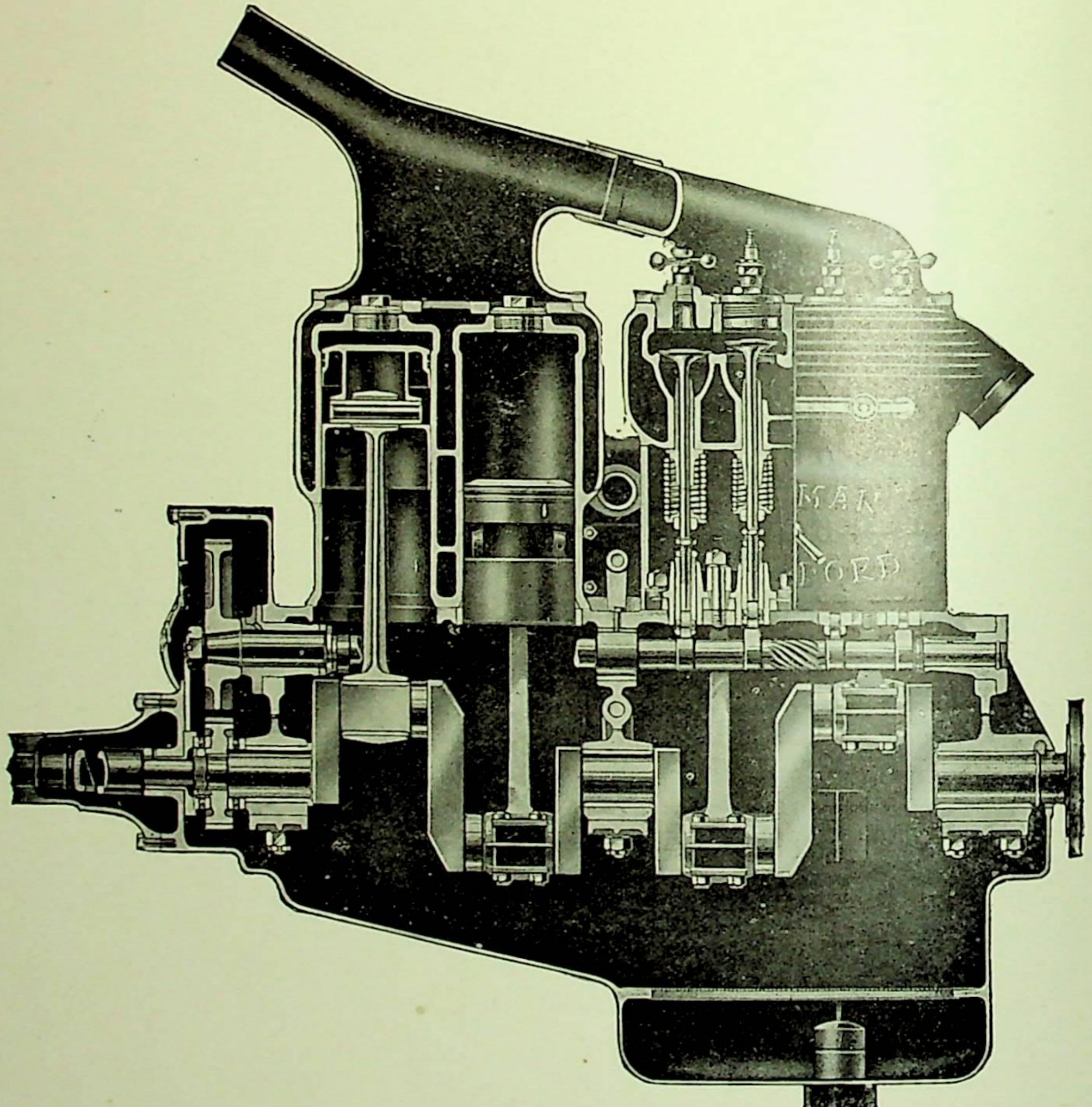


Fig. 117.—The Dorman 40 h.p. Commercial Engine (Side Section).

the interior of the water jackets to be cleaned when they have become coated with fur after lengthy use with hard water.

The crankcase has four short bearer arms which support the engine as a whole in the frame or sub-frame of the chassis. By removing the lower half, which, as already mentioned, performs no duty excepting that of an oil reservoir, the crankshaft bearings, big ends, connecting rods, and pistons can be inspected or removed if need should arise. This is an excellent feature, since there is no need to disturb or to remove the engine as a whole.

The crankshaft, made from a forging of high-tensile steel, is drilled for forced lubrication, and is both stiff and well balanced. It is supported in the upper portion of the crankcase by three die-cast white-metal bearings of ample proportions for durability.

The camshaft is on the left-hand side, and has cams integral with the shaft. Three bearings of large area support the camshaft and are provided for in the lubrication system by having a trough over each one for the reception of oil thrown from the big ends. The crankshaft drive is by means of helical gearing of large tooth surfaces and wheel diameter, the half-speed wheel meshing directly with the driving pinion near the front end of the crankshaft.

In this engine the valves are operated by hollow, roller-ended tappets, having adjustable heads and working in detachable guides.

Throughout this engine is characterised by the ample wearing surfaces of the sliding and rotating parts, in the sturdiness of the design of these, and in the accessibility to the parts requiring periodical attention. It is chiefly in this connection that the heavy engine differs from its smaller confrères.

The pistons for this engine are similar to those shown in Figs. 54 and 55, and are of the Zephyr all-steel type, with two rings above the gudgeon pin.

Fig. 75 illustrates the connecting rods of this engine, the left-hand view showing a rod dissembled. The steel cap, and the two die-cast white-metal liners (held in place in the right-hand view by four bolts, nuts, and lock-nuts) are shown partly detached from the connecting rod. In this design of piston and connecting rod the short, hollow gudgeon pin is fixed in the piston bosses by split pins, and the small end of the connecting rod, which is provided with a phosphor-bronze bush, rocks on the gudgeon pin.

The lubrication system of this engine is of the high-pressure oil-feed type. Oil under pressure is forced to the main journal bearings of the crankshaft and thence to the big-end bearings. For replenishing the sump, oil is introduced through the spring-lid top of a horizontal column forming fan support. The whole of the sump, which is illustrated in Fig. 117, is covered by a detachable gauze strainer of large area. This gauze can be removed through one of the large hand-holes provided in the lower half of the crankcase. A submerged gear-type of pump is employed, such that it is always immersed in the oil sump, and it is driven by means of helical gearing from the camshaft. From the oil delivery an outside pipe leads to a T-piece on the side of the crankcase, one branch being provided for a pipe to a dashboard gauge, or indicator, and the other connected to an internal distribution pipe conveying oil to each of the three crankshaft bearings and to a pressure-relief valve which delivers the surplus oil directly on to the helical timing gears. The sectional view in Fig. 116 shows the oil-level indicating float on the left-hand side, the oil pump on the right and the oil sump.

Fig. 118 shows an outside view of the A.E.C.[1] 45 h.p. heavy-vehicle type engine of 120 mm. bore and 150 mm. stroke. This engine gives 22·5 b.h.p. at 500 r.p.m., and 50 b.h.p. at 1,200 r.p.m. It is of the side-by-side valve type, with the four cylinders cast in pairs in grey cast iron.

The aluminium crankcase is extended below the crankshaft centre in order to provide stiffness, and carries the three-bearing crankshaft running in white-metal-lined bronze bearings, which are mounted in registered aluminium caps with steel stiffening plates. The main bearing bolts are screwed into the crankcase from below, and continue through the case, being finally locked by nuts at the upper end.

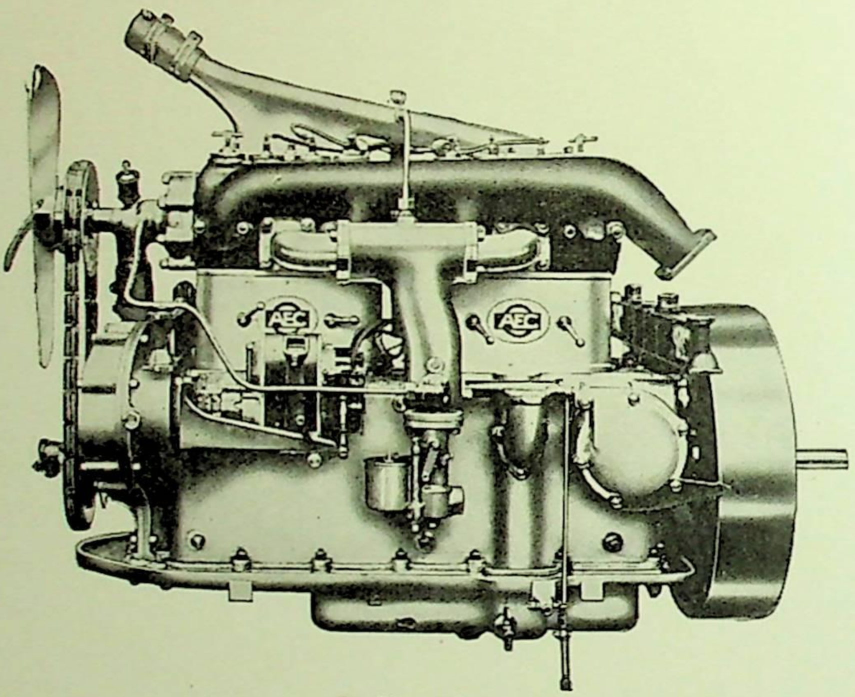


Fig. 118.

The pistons in this engine are of cast iron, with three rings above and a scraper ring below. Below the third ring from the top, an oil relief ring is machined in the piston with suitable oil-escape holes. The drive to the camshaft is by means of a wide silent chain; the drive is a triangular one, the chain passing over the magneto-driving pinion and crankshaft pinion. The magneto is so arranged that it can slide on its platform in order to take up any slack in the chain.

The engine is arranged to be suspended in the chassis by the two overhead stamped-steel cross members, shown in Fig. 118. The front member is attached to the crankcase by an extension of the front main bearing bolts, and the rear member by two bolts passing through the main crankcase casing at its extreme width.

This engine has a hot water-jacketed induction pipe (which can be seen in the centre of the illustration) and is provided with a governing mechanism which prevents the engine speed from exceeding 1,350 r.p.m. (corresponding with a road speed of 20 m.p.h.). A heavy flywheel is provided, to ensure even running at low engine speeds.

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[1] A considerable number of London motor buses are fitted with the engines of this company.

THE SIX-CYLINDER ENGINE

In this engine type, the cylinders are vertical, and the usual practice now is to cast these in groups of two or of three.

The crankshaft is of the six-throw type, with consecutive cranks, on the one side, at an angle of 120°. The two sides (of three cranks each) are symmetrical, so that one-half is the "looking-glass image" of the other; thus the two centre cranks are in one and the same position.

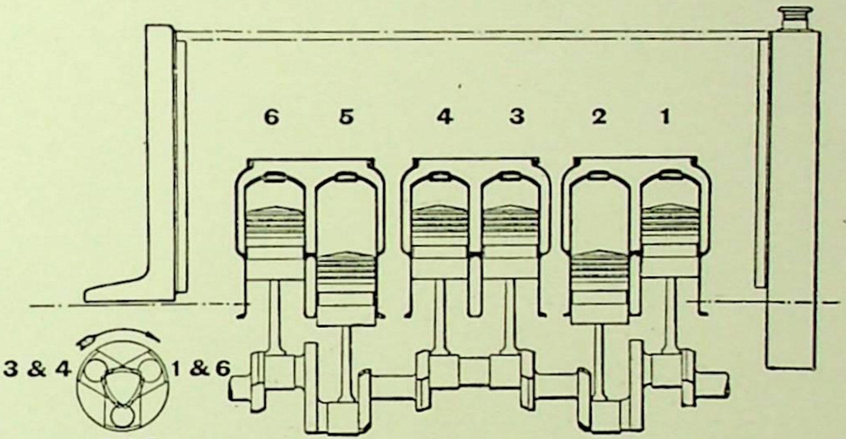

Fig. 119.—The Six-cylinder Vertical Engine.

Fig. 119 is a diagrammatic sketch of the arrangement of the six-cylinder engine, the crank positions for the numbered cylinders being shown in the lower left-hand diagram.

If one imagines this engine to be divided centrally into two three-cylinder engines, then each of the latter corresponds to a type which was at one time quite popular in this country. This type of engine gave a very good balance as far as the vertical reciprocating forces were concerned, and also for the secondary ones, but owing to their lines of action being separated, unbalanced rocking couples occurred. It will now be evident that as the six-cylinder engine consists of two *opposed* sets of three cylinders each, these rocking couples will neutralise one another, and the engine will be in almost perfect balance. As compared with the four-cylinder engine, with its unbalanced

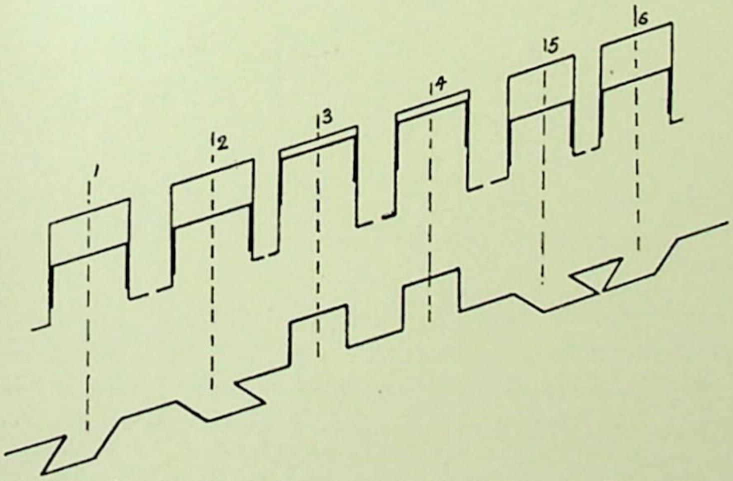

Fig. 120.—Showing Arrangement of Cranks in the Six-cylinder Engine.

secondary forces, the six-cylinder type is undoubtedly superior. There is only a very small unbalanced force, known as the *sixth harmonic*, which vibrates at six times the engine speed. However, it is so small in practice that its effect is hardly discernible, and it certainly would not pay to attempt to neutralise it, as in the case of the Lanchester four-cylinder engine harmonic balancer gear.

Fig. 120 is a perspective line diagram of a six-cylinder engine crankshaft, with the centre lines of the cylinders shown dotted. The six-cylinder engine is longer than the four, but possesses the advantages of better balance, smoother running, better torque (a smaller flywheel being required), and a wider speed range on top gear for the car thus fitted. It is, on the other hand, more expensive to manufacture and more troublesome to overhaul, on account of its 50 per cent. greater number of cylinder parts.

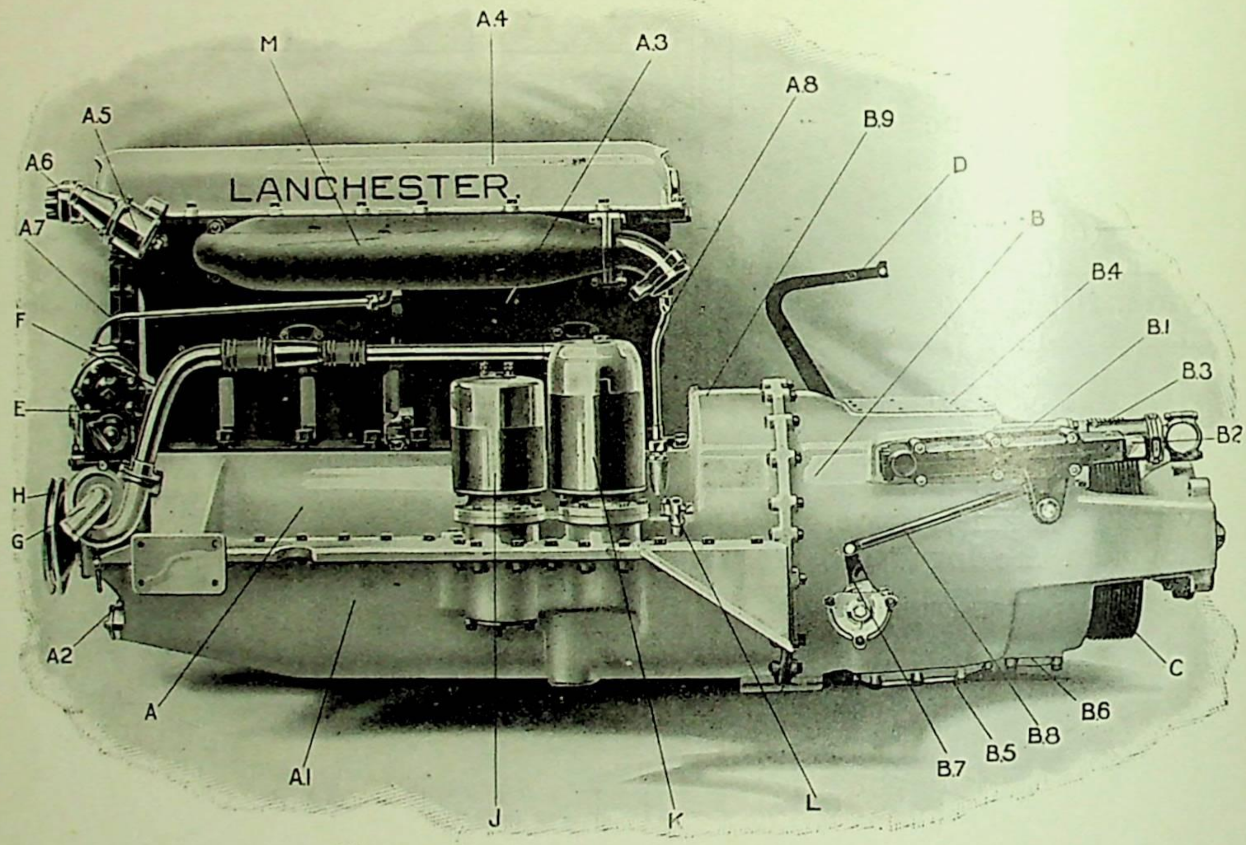

Fig. 121.—The Lanchester Six-cylinder Engine.

The firing order of the cylinders, as numbered in Fig. 119, is as follows, namely: 1, 5, 3, 6, 2, 4. In the case of a left-hand engine the firing order is 1, 4, 2, 6, 3, 5.

The Lanchester 40 h.p. six-cylinder car engine is illustrated in outside side elevation in Fig. 121 and in section in Fig. 122 on Plate facing page

104. For the sake of clearness the various parts and components have been annotated, the corresponding parts being as follows in the case of Fig. 121, namely:

A, Crankcase
A1, Crankpan
A2, Crankpan Oil-gauge Cover
A3, Cylinders
A4, Valve Cover
A5, Thermostat
A6, Distributor
A7, Oil Pipe
A8, Oil Pipe
B, Change-speed Gear Box
B1, Gear Push-bars Housing
B2, Gear Push Bars
B3, Clutch Spring Box
B4, Gear-box Inspection Lid
B5, Gear-box Oil Sump
B6, Gear-box Oil Pump
B7, Clutch Lever
B8, Clutch Adjustment Link
B9, Flywheel Casing
C, Countershaft Brake Drum
D, Clutch Pedal
E, Magneto
F, Water Pipe
G, Water Pump
H, Fan Pulley
J, Starting Motor
K, Generator
L, Oil-pressure Relief Valve
M, Exhaust Manifold

Before proceeding to the corresponding annotation of the various parts in the sectional view, a few particulars of the Lanchester engine will be given.

The engine is of the six-cylinder vertical type previously described, and has a bore and stroke of 102 mm. (4 inches) and 127 mm. (5 inches) respectively. It has overhead valves and camshaft, worm driven, is totally enclosed, and is lubricated by automatic means. High-pressure forced lubrication, spray carburettor and water-jacketed induction pipe, two independent ignition systems, water circulation by centrifugal pump, a large exhaust manifold, and electric starter and generator are designed integral with the engine.

Reference has been made already to the Lanchester engine crankshaft, overhead-valve gear arrangement, and vertical driving shaft. The engine cylinders are in two blocks of three each, in cast iron, the combustion heads and valve chambers being cast integral with the cylinders, thus ensuring a free circulation of water around the most essential parts, namely, the valves and sparking plugs, and eliminating the possibility of water and gas leakage, to which detachable heads are susceptible. The combustion space is of compact, partially spherical form and is machined all over, which ensures very good combustion efficiency and uniformity of cylinder mixture volume. Both crankcase and oil base are of deep section, combining to form a stiff girder-like construction, the strength of which is materially increased by the wide-flanged oil joints. The main bearing brackets are carried by massive webs, cast transversely in the crankcase.

The oil base is provided with a gauge filter of wide area which may be withdrawn through a wide aperture in the crankcase.

Aluminium pistons made by the die-casting processes are employed, these pistons having been adopted after a six-years' duration test. Four piston rings are fitted, one of which is at the bottom of the piston skirt and serves as a scraper ring. The gudgeon pin is tubular and is case-hardened. This pin is of the floating type, the piston bosses being bushed for the purpose. The gudgeon-pin bushes in the piston are held in place by means of two spring circlets which are pressed into annular grooves formed in the bore of the piston bosses. The connecting rods are of the H-section drop-forged type, but machined all over; they are lined with white-metal bearings cast under pressure, direct on to the steel. The small end is bushed with phosphor bronze, and is lubricated by oil which is conducted from the crankpin bearing to it by means of a steel tube secured in the rod before the insertion of the bearings.

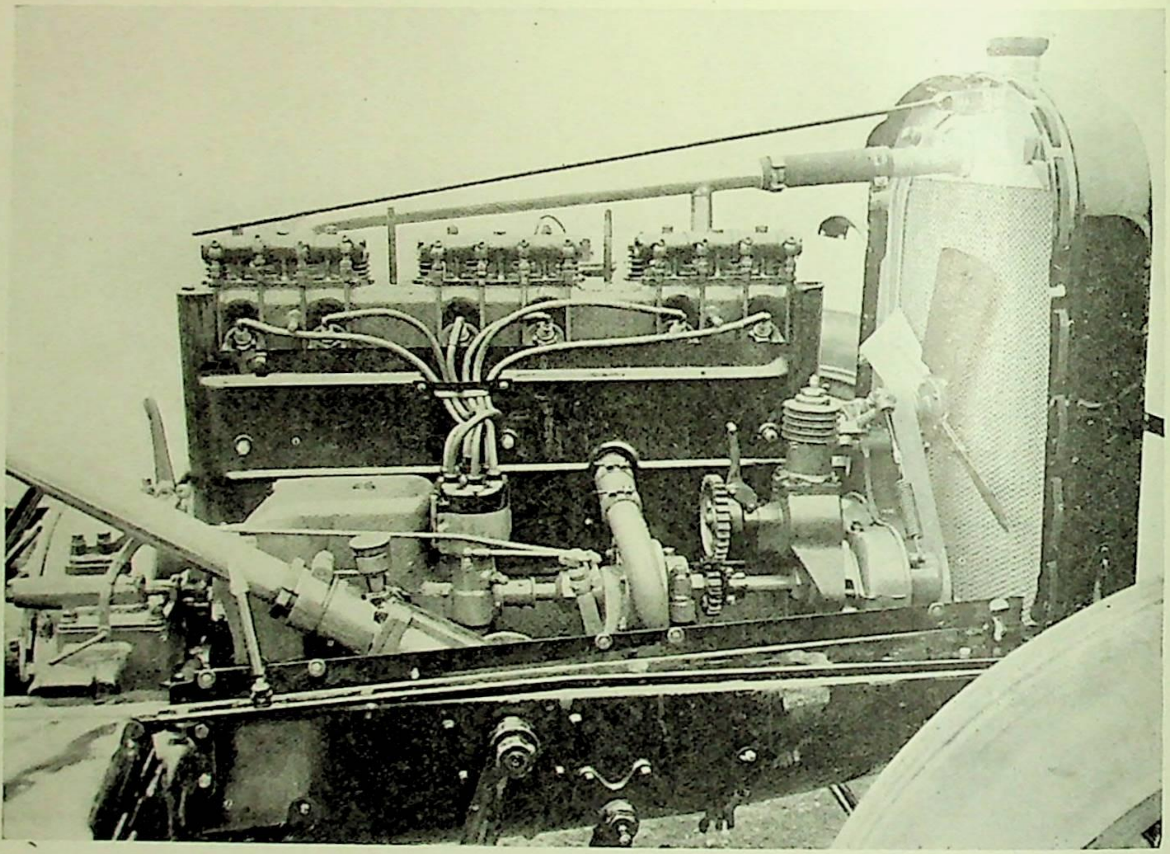

Fig. 123.—The Buick Six-cylinder Engine.

Referring to Fig. 122, which shows the Lanchester engine in section, the key to the annotation employed is given below the diagram and will enable all of the important parts to be traced.

As an example of American six-cylinder engine practice in the ordinary classes of touring car, the Buick engine may be quoted.

This engine, which is shown illustrated in Fig. 123, is of the six-cylinder vertical type with cast-iron monobloc cylinders. The dimensions are 85·7 mm. ($3\frac{3}{8}$ inches) bore and 114·3 mm. ($4\frac{1}{2}$ inches) stroke, the engine giving a nominal b.h.p. of about 27.

Overhead valves, operated by long adjustable push rods are employed, valve covers being fitted.[1] The sparking plugs, which can be observed in the illustration, are placed in an inclined position for accessibility. The hot-water outlets from the two ends of the cylinders lead, by means of a straight pipe, to the upper part of the radiator, the cooling fan for which is driven by a flat belt with a spring-tensioning device.

The accessories of the engine are clearly shown, and comprise a small single-cylinder air pump for the tyres, the water-circulating pump (of the centrifugal type) next to it, the distributor of the Delco high-tension coil system with its leads to the sparking plugs, and the Delco lighting and starting electrical unit.

The lubrication of this engine is by the splash process, with a gear pump for keeping the oil troughs supplied.

THE EIGHT-CYLINDER ENGINE

The eight-cylinder engine is now employed only upon the more expensive cars, and also for racing cars; it stands in a somewhat similar position to the six-cylinder engine that the latter does to the four-cylinder one, in the matter of flexibility and uniformity of torque.

There are two favoured arrangements of this type, namely, the Vee type, in which two rows of four cylinders each are inclined to each other, and the vertical type (known as the *line-ahead* or *straight eight*), in which the eight cylinders are in the same vertical plane, one behind the other.

The ratio of the maximum to the mean torque for this engine is about 1·25, as compared with 1·4 for the six-cylinder and 2·0 for the four-cylinder engines previously considered on pages 89 and 101.

The Vee-type engine can be made of the same overall length as the ordinary four-cylinder vertical type, and a manufacturer of the latter type can often arrange for the eight-cylinder model to employ two sets of four-cylinder parts (e.g. cylinders, pistons, and valves). The crank-case will, of course, be different, as also will the connecting rods (two rods working on one crankpin), and the crankshaft will be much stiffer.

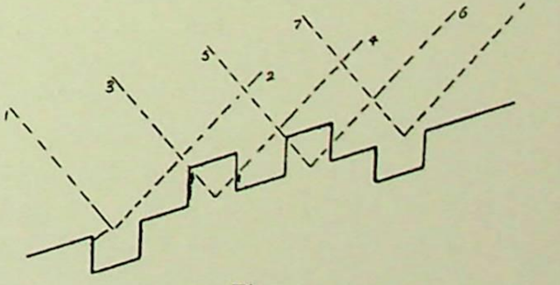

Fig. 124.

Fig. 124 illustrates diagrammatically the crankshaft and the axes of the cylinders in the case of a Vee-type eight-cylinder engine. This type enables about twice the power to be obtained from an engine of the same bonnet space as the four-cylinder type. It is lighter than an eight-cylinder vertical engine of the same cylinder dimensions, and has only one-half the number of crankshaft bearings, besides possessing a stiffer yet lighter crankshaft.

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[1] These are not shown in the illustration, but the two securing bolts can be seen.

The balance of the Vee-type engine is somewhat similar in effect to that of the four, there being a synchronism of the unbalanced secondary forces of total magnitude equal to a four-cylinder engine having the same reciprocating mass, very approximately.

The order of the firing of the cylinders, using the notation of right- and left-hand rows, each numbered from the front end from 1 to 4 respectively, is as follows, namely: 1R, 4L, 3R, 2L, 4R, 1L, 2R, 3L. There is, however, an alternative firing order which can be used, namely: 1R, 4L, 2R, 3L, 4R, 1L, 3R, 2L.

The angle between the cylinders is usually 90°, but smaller angles down to 50° have been employed.

The eight-cylinder vertical engine enjoyed a period of popularity in 1920 and 1921, and cars fitted with this type of engine performed very creditably in car races, including the Grand Prix in 1921.

It is rather from the racing-engine viewpoint that the straight-eight engine has been manufactured, since it possesses certain advantages in this respect. Briefly, this type enables the highest power output, with a high degree of reliability, to be obtained.

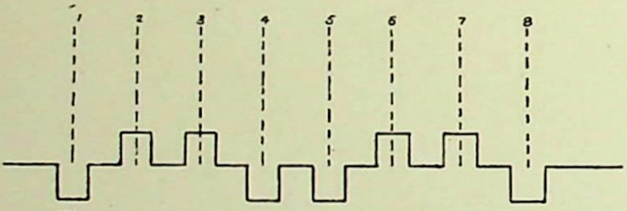


Fig. 125.—The "Straight Eight" Crankshaft.

The absence of a flywheel, owing to the very even torque, results in quicker acceleration and deceleration. The centre of gravity can be made lower, owing to the shorter stroke, whilst the balance of this type is superior to that of a four-cylinder engine.

The reduction in size of the engine cylinders can be shown to result in an increase in the engine speed and output per unit volume, and it is a well-known fact that whereas with a large piston a petrol engine's maximum speed will seldom reach 2,700 r.p.m., yet with a small piston it will exceed 3,500 r.p.m.

Compared with the eight-cylinder Vee-type engine, it is possible to allow greater main-bearing and big-end bearing area, and thus to reduce the bearing pressures—an important point in high-speed engines—so that the straight-eight engine will run for longer periods at high speeds with a less likelihood of bearings giving out.

THE TWELVE-CYLINDER ENGINE

The popularity of the eight-cylinder Vee-type led to the introduction of the twelve-cylinder Vee type; but it is due chiefly to the development of the aircraft engine, notably the Rolls-Royce and Sunbeam engines, that this type has found favour.

There are one or two cars which still employ this type of engine, including the Packard twin six, but these do not appear to be growing in numbers, owing, probably, to the development of types with a smaller number of cylinders.

The twelve-cylinder Vee type consists of two sets of six cylinders, inclined at a small angle to each other (from 60° to 90°), and employing the same crankshaft arrangement as in the six-cylinder type. As would be anticipated, this type gives extraordinarily smooth running, the torque being practically uniform all the time, and as a result the running properties of the car on the road are excellent as regards acceleration, flexibility, and speed range on top gear. It is questionable, however, whether this multiplicity of working parts fully justifies the results obtained, in view of the excellent balance and torque of the eight- and six-cylinder engines.

The connecting rods in this engine are usually forked on the one side and plain on the other, as two big ends have to be accommodated on the same crank journal.

Fig. 126 is a front sectional view of a twelve-cylinder engine, with the cylinders at 45°, the cylinders being separate and of machined steel, with welded water jackets. Overhead valves are employed, worked by two overhead camshafts, driven by vertical shafts and gearing from a gear wheel on the crankshaft. Twin magnetos and twin carburettors are employed, the latter being placed in the Vee formed between the cylinders. The water outlets at the top in the centre and bottom on the outsiders can be seen in the section shown. The exhaust pipes are separate for each cylinder set, and lead to its own silencer, of which there is one upon each side.

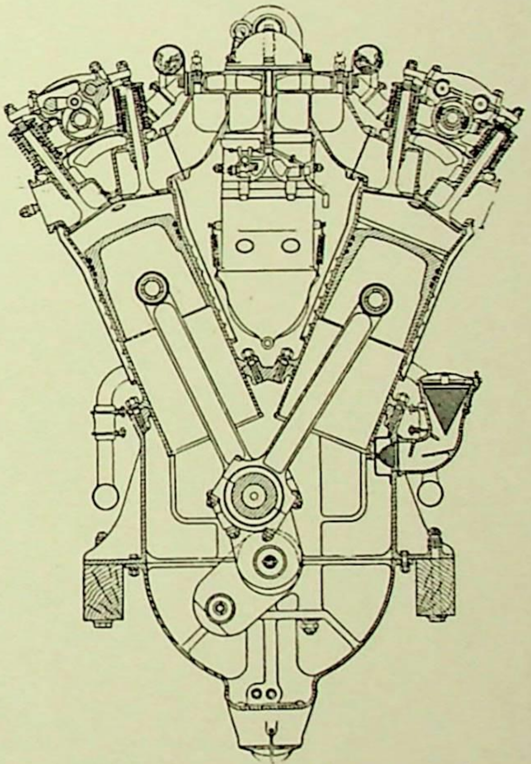


Fig. 126.—Twelve-cylinder Vee-type Engine (End Section).

Frequently, in the case of smaller car engines, the one cylinder block contains one-half of the cylinders, so that there are virtually two sets of six-cylinder engines working upon the same crankshaft.

The firing order usually adopted is as follows, namely: 1R, 6L, 4R, 3L, 2R, 5L, 6R, 1L, 3R, 4L, 5R, 2L; but there are two alternatives to this which will not, however, be mentioned here.

In the case of the Packard twin-six engine shown in Fig. 127, the cylinders are in two blocks of six, each at 60° to each other, the bore and stroke being 76·2 mm. (3 inches) and 127·0 mm. (5 inches) respectively. The cylinders are of cast iron, and fitted with detachable heads to facilitate cleaning.

The horse-power of this engine is nominally 43·2, but the results of tests show that it can develop over 75 h.p.

The crankcase is of aluminium alloy, and has four-point attachment to the engine frame; two of the supports are shown in the illustration. A three-bearing crankshaft of $2\frac{1}{4}$ inches diameter is fitted.

The ignition arrangement consists of a dynamo, storage battery, and Delco high-tension distributor system, and there is a separate starting motor.

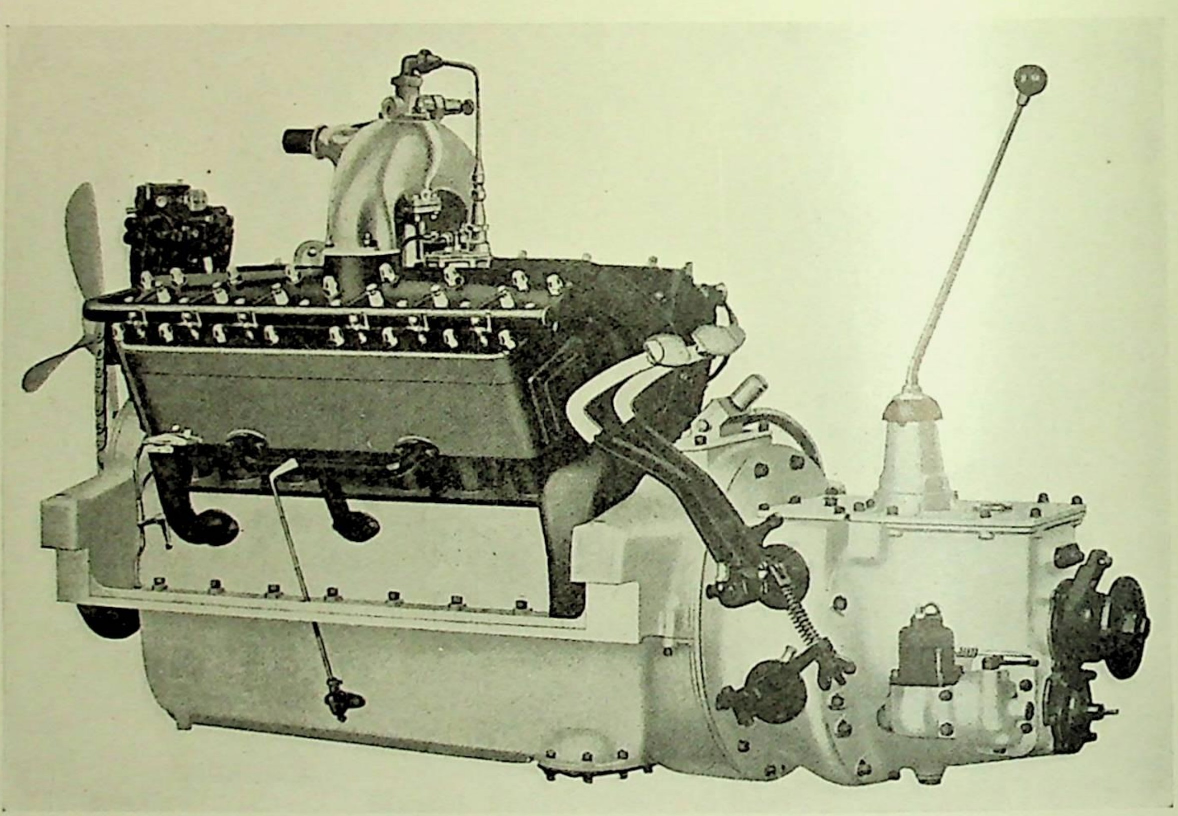


Fig. 127.—The Packard Twin-six Engine.

The illustration shows also the unit gear box attached, together with the left-hand brake and clutch pedals, and the central change type of gear lever so popular in American cars, and which is now finding favour in this country.

The lubrication system is a pressure-fed one, all the principal bearings being fed with lubricating oil at a pressure of from 20 to 25 lb. per sq. inch for normal running, the pressure increasing with the power output.

SPECIAL TYPES OF ENGINE

Although the poppet-valve type of engine is almost universally employed, there are other types of engine working upon the two- or four-stroke principles, which differ from the former in the charge induction and exhaust arrangements. Ever since the practical adoption of the poppet-valvetype, attempts have been made to avoid this apparently crude valve system and to utilise other means for regulating the inlet and the exhaust processes, such as by the use of rotating valves, slides, or sleeves. A few only of these have met with any real success, although some hundreds of inventions have been patented, or applied for.

Generally speaking, the principal advantages of the poppet or mushroom-head type of valve are: (1) its simplicity; (2) the fact that it is protected against the effect of the exploding gases, that is to say, the working surfaces are unexposed except for the valve seating during the exhaust period; and (3) the fact that there are no sliding surfaces regulating the processes.

The principal disadvantages of the poppet-valve type are: (1) its noisiness; (2) the fact that it frequently gives a bad shape of combustion chamber; (3) the occasional adjustment of the clearances which is required; and (4) frequent bouncing of the valves at very high speeds, and spring periods affecting their correct functioning, and occasionally causing fracture.

The most notable departures from the practice of the poppet valve are the Knight double-sleeve valve (used on the Daimler, Minerva, and other cars), the Burt-M'Collum single-sleeve valve (used in the Argyll and Barr and Stroud engines), the Itala rotary valve, the Darracq, and the concentric or "cuff" valve. Of these, the double- and the single-sleeve valve types alone appear to have held their own, if the two-stroke type of *valveless* engine be considered apart for the moment. As the Itala and rocking-valve types are still of considerable interest, it is proposed to mention them briefly in the present considerations, and in view of the possibility of similar devices arising in the future.

THE SINGLE-SLEEVE VALVE ENGINE

The success attending the double-sleeve valve Knight engine no doubt led to the adoption of the single-sleeve engine made under the Burt and M'Collum patents. This latter type has met with equal success in both motor-cycle and car engines.

The single-sleeve engine fitted to Barr and Stroud motor-cycles was adopted only after a considerable amount of experiment, as the air-cooled engine presents special problems in the application of this principle. This engine, which is shown in sectional illustration in

Figs. 128 and 129, will serve to explain the principle of the sleeve-valve action.

It will be observed that the piston works in a concentric sleeve, extending down the whole length of the cylinder, which is provided with ports in the vicinity of the inlet and exhaust manifolds. This sleeve, which constitutes the valve system, is operated at one-half the speed of the crankshaft through a two-to-one reduction gear. Its motion is of a combined reciprocal and rotary nature, so that any point on its surface describes an ellipse in completing a cycle of movement. It will be evident that with this construction the cylinder and combustion chamber can be made without pockets, and the latter of a shape approximating to theoretical ideals.

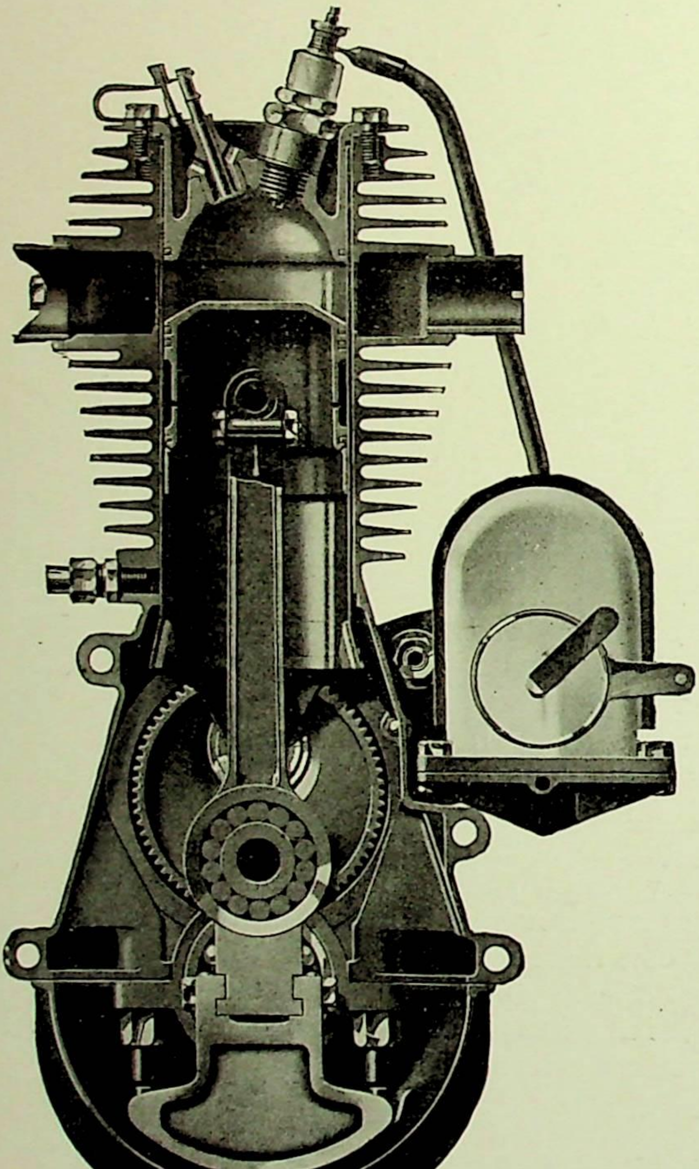


Fig. 128.—The Barr and Stroud Engine.

The action of the sleeve valve is as follows: The sleeve, working at half-engine speed, makes one complete travel in its elliptical path for every two revolutions of the crankshaft. The actuating pin formed on the sleeve (at its lower end) is driven round in a nearly circular path by the inner member of the ball-and-socket joint located within the half-time wheel. As the pin retains its horizontal position during the entire stroke, the vertical or reciprocating movement of the sleeve is equal to the diameter of the circle described by the centre of the bearing (in this case 36·5 mm.); but the rotating movement of the sleeve is less, owing to its outer surface being nearer the centre of rotation, namely, its own axis, than is the actuating pin, so that the net rotary movement is only 20·5 mm. Thus the resultant path of any point on the surface of the sleeve is very nearly an ellipse with its longer axis vertical, and is anti-clockwise in direction as viewed from the outside.

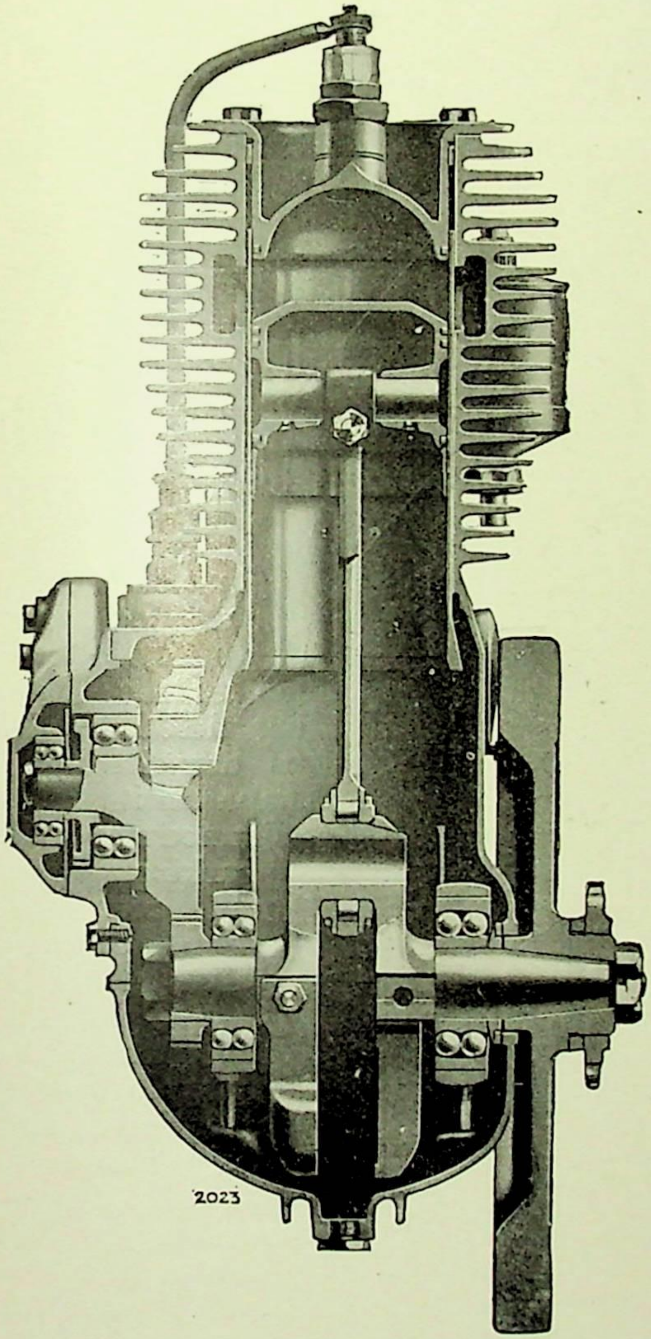


Fig. 129.—The Barr and Stroud Engine.

Fig. 130 is a cross-sectional plan view of the cylinder through the ports and passages, the exhaust ports being shown open and the inlets closed; the solid portions of the sleeve are shown in black.

Fig. 131 is a port-opening diagram, showing the relative positions of the sleeve and cylinder during a complete cycle of operations. The relative positions of the cylinder ports and the sleeve ports at six points in the cycle of operations are shown, namely:

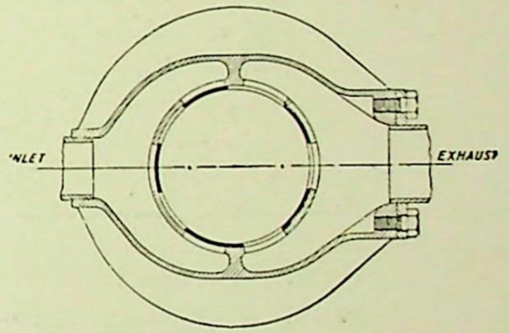


Fig. 130.

(1) Induction ports beginning to open.

(2) Induction ports fully open.

(3) Induction ports closed, compression commencing.

(4) Top of compression stroke, firing occurs.

(5) Exhaust ports beginning to open.

(6) Exhaust ports fully open.

It should be explained that each set of shaded horizontal diagrams

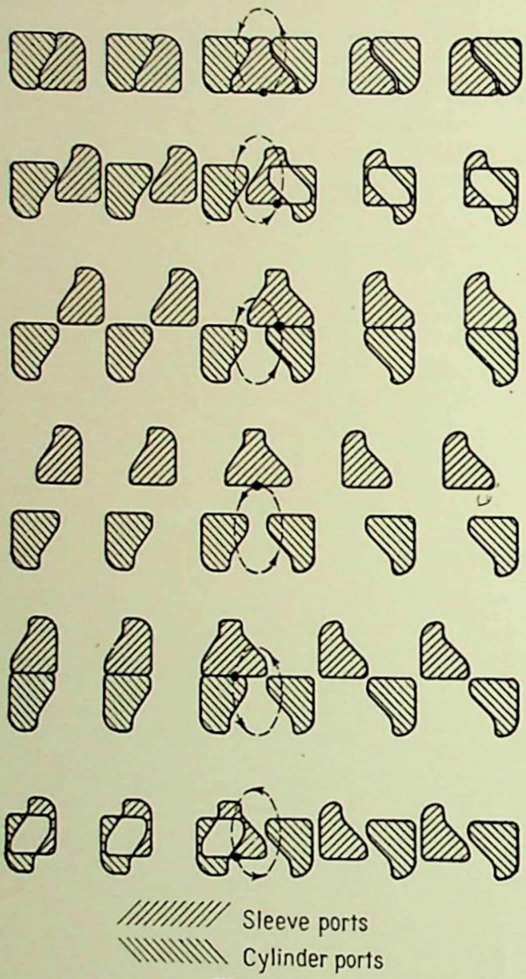


Fig. 131.

represents the development, or flattened-out sleeve with three exhaust ports and three inlets, the centre one being common for the sleeve. The dotted ellipse represents the path of a point on the sleeve.

The dimensions of the engine illustrated are 70 mm. bore and 90·5 mm. stroke, the compression ratio being 4.

Fig. 132 is a reproduction of a power-output curve of one of these engines, and the results show that 4 b.h.p. is obtained at 2,000 r.p.m.; whilst at 4,000 r.p.m., 7 b.h.p. is developed, which is a high power output from an engine of only 350 c.c. capacity.

The arrangement of the combustion head in this engine is of some interest. From Figs. 128 and 129 it will be noticed that the cylindrical head acts as a guide for the sleeve, a piston ring being fitted in the head. The head is detachable, is hemispherical in shape on its lower surface, and accommodates the sparking plug and compression-release valve.

The lubrication of the sleeve calls for special attention in this design, and is provided for by leading the oil to a point in the cylinder near the lower end of the sleeve (see Fig. 128), whence it is distributed by the oval motion of the latter and

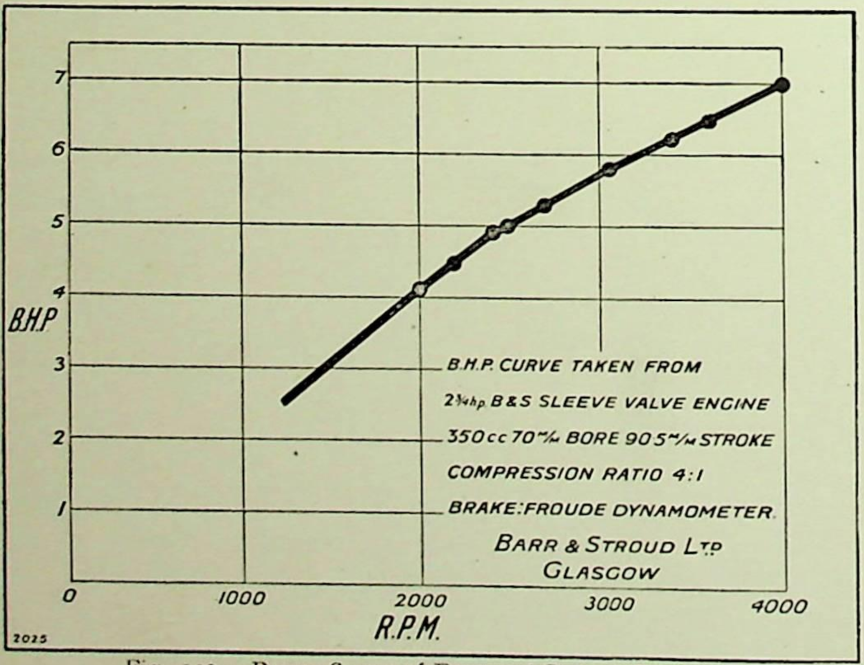


Fig. 132.—Power Curve of Barr and Stroud Engine.

spreads all over the surface. The oil which drains into the crankcase is splashed on to the bearings, and tests show that this system is quite satisfactory.

The crankshaft is made from a solid high-tensile steel forging and is very short and rigid. Ball bearings of the double-row type are fitted to the main journals and to the sleeve-actuating gear. The crankpin houses the roller bearings for the big end. In order to assemble these roller bearings, the balance weights are detached, and the bare connecting rod is threaded around the crankshaft, after which the rollers are placed in position, where they are retained by split side-plates of bronze, held together by means of steel locking rings.

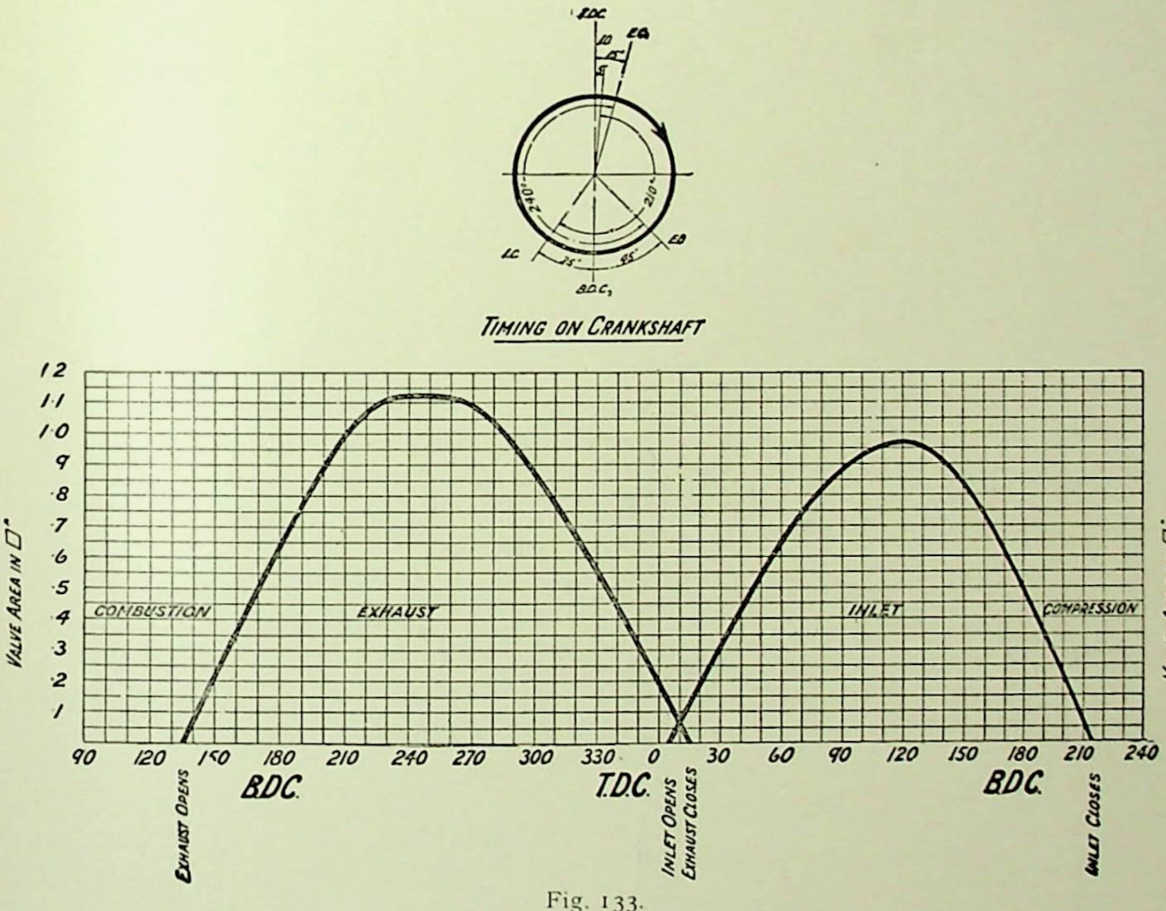


Fig. 133.

The timing diagram of this engine is shown in Fig. 133, the amounts of inlet and exhaust port openings (in sq. inches) being shown on a crank-angle base. It will be noticed that the exhaust-valve opening is appreciably more than the inlet opening, and that the opening of the exhaust and closing of the inlet are performed with special rapidity, a large aperture being uncovered for the greater portion of the available time.

This design of engine scores over the poppet-valve type in the above respect, in its simplicity and absence of hammering and noisy valves and valve springs, and in its better combustion efficiency. Although the sleeve is heavier than the poppet valves which it replaces, less power is probably absorbed owing to its small continuous motion.

The action of the water-cooled type of single-sleeve valve engine shown in Fig. 134 is exactly similar to that described, the only differences between the two being in the design of the water jackets and ports, and in the multiplicity of cylinders, valve-operating mechanism (seen on the right), and lubrication arrangements. In this case the valve-operating shaft B is geared to the crankshaft by a silent chain and revolves at the same speed. On this shaft opposite each cylinder is a small worm which engages with a disc wheel revolving at one-half engine speed; this disc actuates the sleeve and imparts to it the required oval motion.

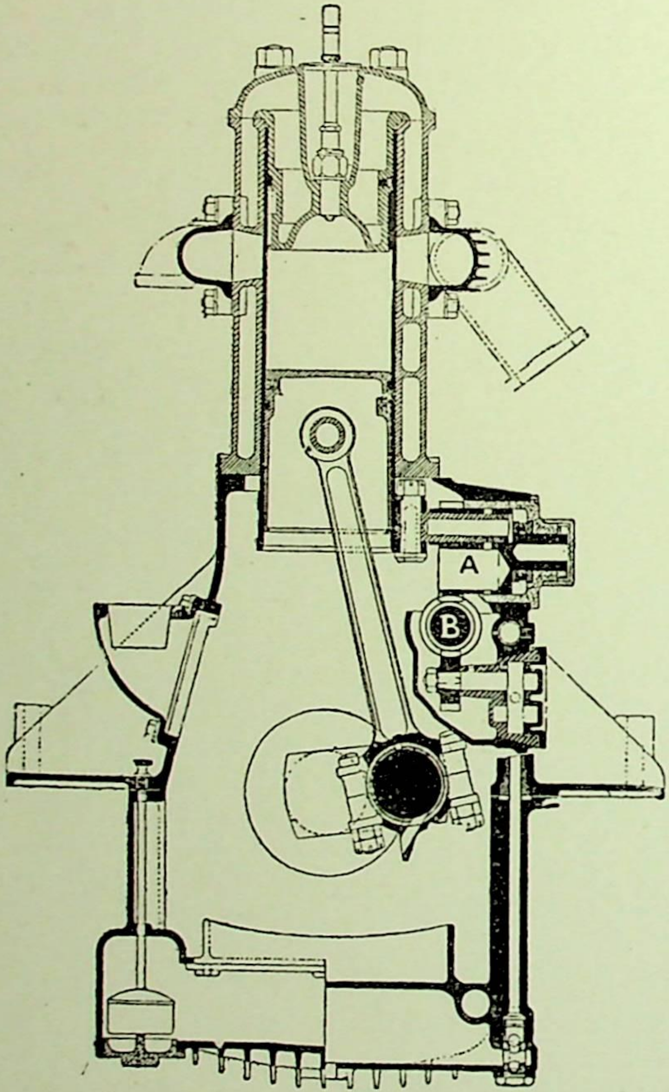


Fig. 134.—Section of Sleeve-valve Engine.

In connection with the latter, it should be emphasised that a circular or oval motion between two sliding surfaces produces appreciably less friction, and better distribution of the lubricant, than a mere rectilinear sliding motion. The wear in this type of engine is very small, and hence the life is exceedingly long. When replacements are required (after long running), it is generally found that the sleeve and piston only require renewal.

The silent running of this type of engine is most noticeable, especially at high speeds and loads, provided, of course, that an efficient silencer is fitted. This silence is not, however, obtained at the expense of power, for, as will be seen, also, when the Daimler type of sleeve-valve engine is described, it is remarkably efficient in the matter of power output for a given cylinder size.

THE DAIMLER SLEEVE-VALVE ENGINE

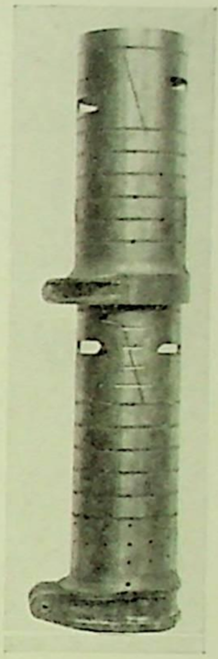


Fig. 135.

Mr. Knight, in 1905, took out a patent (No. 14,729) for a sleeve-valve type of engine on the four-cycle principle; and later, in 1908, Messrs. The Daimler Company, of Coventry, after a series of exhaustive tests and investigations, decided to abandon the poppet-valve type of engine in favour of this sleeve-valve type. This change over, by a leading automobile concern, was a sudden and bold stroke of engineering policy, but the results obtained since have fully justified the decisions.

Before describing the latest types of this engine, its principles of working will be described.

The Daimler engine is unique in respect to its absence of the usual poppet valves, and employs for the admission and exhaust of the gases two concentric vertical sleeves provided with suitable ports in their upper ends. The piston reciprocates inside the inner sleeve. The illustration shown in Fig. 135 will give some idea as to how the lower inner sleeve slides within the upper outer sleeve. It also shows very clearly the peculiar system of grooves and oil holes on the surfaces of the sleeves for the purpose of distributing the lubricating oil over the sliding surfaces. The stiffened bosses at the lower ends of the sleeves carry the pins for the operating mechanism, which is shown in Fig. 136. The inner sleeve is there shown within the outer, and the two sets of ports are shown about one-half open.

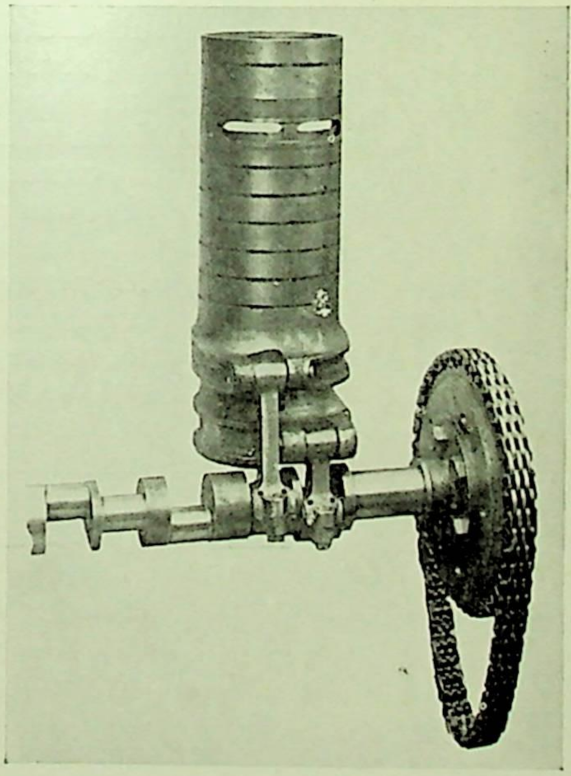


Fig. 136.—The Daimler Sleeve Valves and Valve Gear.

It will also be observed that the sleeves are reciprocated up-and-down by means of short connecting rods operated off a crankshaft of small throw. The latter is driven at one-half engine speed by means of a silent chain directly from the main crankshaft of the engine, which in the present case is of the six-cylinder type. Fig. 137 shows one of the cylinder blocks of two cylinders, with a pair of sleeves in place. The cylinder ports can be clearly seen, and also the sleeve ports. A notable feature of this engine is its external clearness of design, due to the absence of the usual poppet-valve mechanism.

Having described the principal components of the sleeve-valve system, we shall next consider its mode of operation ; and shall refer to Fig. 138 for this purpose.

In (1) the complete cylinder with its sleeves, water jackets, and combustion head is shown in section, together with the piston (in section) and the connecting rods for operating the two sleeves ; the directions of rotation are indicated by the arrows. It will be observed that in (1) the outer sleeve is moving downward, whilst the inner sleeve is moving upward, the ports communicating with the inlet being about to open. As the piston descends on its next stroke the inlet ports will have opened and the mixture will be drawn into the cylinder until, when nearing the end of its stroke, the sleeves will have closed the inlet ports. In the first illustration the exhaust port is shown on the point of closing, as this is the end of the exhaust stroke preceding the suction stroke.

In Fig. 138 (2) both sets of ports are shown closed, and the mixture is being compressed on the compression stroke, the piston (not shown) moving upwards. Fig. 138 (3) shows the ending of the ensuing firing stroke when the exhaust port begins to open, the inlet port being still closed. Fig. 138 (4) shows the end of the exhaust stroke again, with the inlet just on the point of opening and the exhaust just closing. The cycle of operations is thus complete.

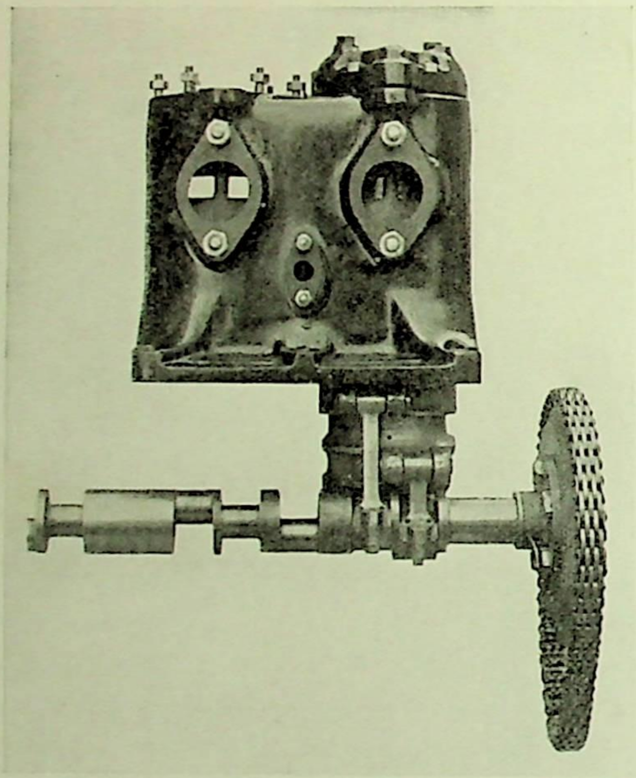


Fig. 137.

Referring to Fig. 138 (1), it will be seen that at their upper ends the sleeves slide in an annular space between the cylinder wall proper and an inner detachable cylinder head of very nearly hemispherical shape on the lower side. This head is amply water jacketed, and is provided with three piston rings to prevent leakage of the gases ; it also accommodates the centrally placed sparking plug. The form of combustion chamber obtained in this case, namely, an approximately spherical one, is excellent and conforms to theoretical requirements in giving maximum cooling area for a given volume. It has been shown that with such a shape there is less tendency to "pinking" or detonation when higher compressions are used, and when heavier grades of petrol or paraffin are employed.

An external view of the Daimler six-cylinder 45 h.p. sleeve-valve engine is given in Fig. 139. This engine is, however, manufactured in the 12, 16, 21, and 30 h.p. sizes for different types of cars.

The illustration shows the four-bladed aluminium cooling fan driven by a flat belt off the lower pulley, which is formed on the

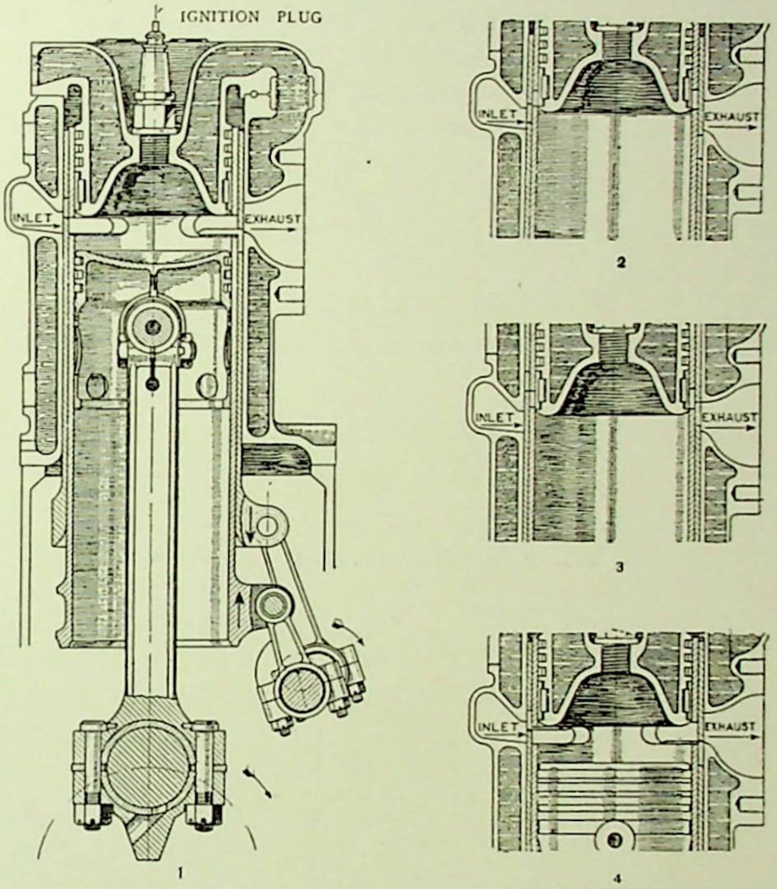


Fig. 138.—Illustrating the Action of the Daimler Sleeve-valve Engine.

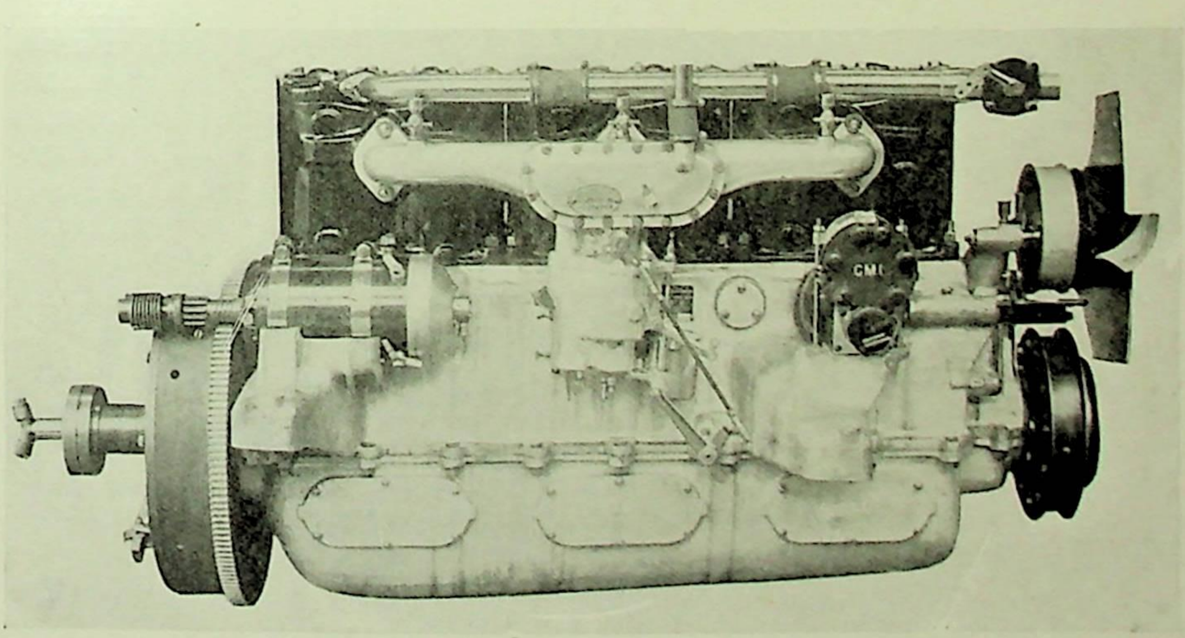


Fig. 139.—The 45 h.p. Daimler Sleeve-valve Engine.

vibration damper, a device illustrated in Fig. 140 and described below. The carburettor is shown in the centre, with the hot-water-jacketed induction pipe immediately above it ; below can be seen a pivoted elbow lever connected by an inclined rod to the throttle of the carburettor. This is for the purpose of controlling the amount of lubrication in the engine according to the engine load, which, in its turn, is dependent upon the throttle opening. This feature is referred to in more detail in the chapter on lubrication.

In the illustration in question the flywheel and clutch, and the starting motor with its pinion, can be seen on the left-hand side.

THE LANCHESTER VIBRATION DAMPER

The Daimler engines are fitted with a device, invented by Mr. F. W. Lanchester, for reducing the vibrations due to the variation of the torque on the crankshaft. It is well known that if a spiral spring in tension receives a regular series of impulses it will be set in a violent state of vibration when these impulses occur at certain speeds, or rather intervals of time. The crankshaft of a petrol engine may be regarded as a spring, but under torsion, and the intermittent torques due to the firing strokes to the impulses, so that the general result is to tend to make the crankshaft vibrate or oscillate about its axis, in torsion. This effect is, of course, superposed upon its ordinary rotation. As the torque variations occur at high frequency, so do the vibrations of the crankshaft, as a rule, but the actual frequency depends upon the design of the shaft and the material used.

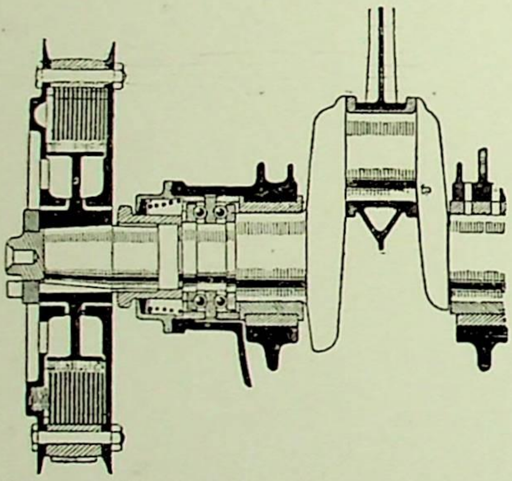


Fig. 140.—The Lanchester Torsion Vibration Damper.

It is only when the firing impulses happen to coincide with the natural frequency of vibration of the crankshaft that the vibrations set up are of serious importance. Thus at certain speeds, which are harmonics as a rule, the whole engine exhibits a tremor, and in the case of closed-in type bodies, a drumming effect often occurs. These speeds are usually spoken of as the "critical speeds" of the engine, and it is the aim of the designer to make the engine speeds corresponding to the ordinary driving speeds of the car different from the critical speeds.

The Lanchester vibration damper, shown in section in Fig. 140, was designed for the purpose of damping out the torsional vibrations. It is embodied in the pulley which drives the fan (see Fig. 139), and therefore serves a double purpose. Instead of keying the fan-belt pulley itself direct to the crankshaft, it is driven through a multiple-disc clutch, composed of thin steel plates engaging alternatively with the crankshaft and the fan-pulley drum. The frictional grip between the plates is obtained by the aid of springs, and the whole mechanism is enclosed in an oil bath. The crankshaft vibrations are thus damped out by the friction of the steel plates and the oil-damping action.

ROTARY VALVE ENGINES

A type of petrol engine which differs in its valve operation from either the sleeve or the poppet-valve engine is that which employs a rotating cylinder or disc, with a suitable port, or ports, cut in it for controlling the inlet and exhaust operations. The general principle of this type of engine will be evident from the diagrammatic view shown in Fig. 141 of a rotary valve in the head of a cylinder. There the inlet port is shown to the right and the exhaust port to the left of the cylinder axis, and a cylinder which closely fits the cylindrical cavity above is made to rotate at a suitable speed, namely, one revolution to two of the crankshaft. The cavity in this rotating cylinder, or valve, alternately connects the inlet and the exhaust ports with the combustion head of the cylinder, whilst during the compression period it seals off the mixture in the cylinder. In this way the ordinary operations of admission, compression, and exhaust are performed.

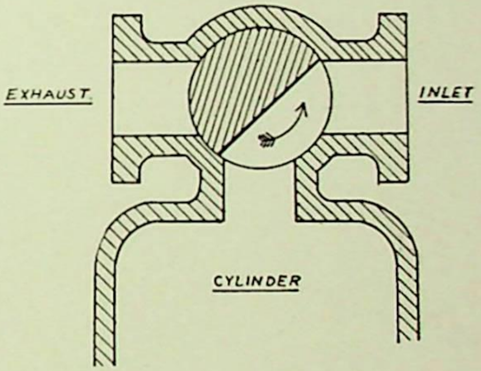


Fig. 141.

Numerous types of rotary valve and piston engines have been devised, patented, or tested in the past, of which the Henriod, Itala, and Darracq types are the more noteworthy examples. The two latter engines had a certain vogue a few years ago, and it may therefore be of interest to describe one of these, before inquiring into the reasons for the lack of continued success, and of the difficulties involved in the design and operation of these types.

Fig. 142 illustrates the Itala rotary-valve engine as fitted to Itala cars a few years back ; four sectional views are shown, each corresponding to a certain phase in the cycle of operations. The cut-away vertical cylinder seen on the right-hand side is given a rotary movement about its axis by means of the vertical shaft shown. This shaft is driven by means of a worm-gear from a gear-wheel meshing with another affixed to the crankshaft. Each rotary valve, however, controls the admission, compression, and exhaust operations of two cylinders, as will perhaps be clear from the sectional views given in Fig. 143 for the four cylinders,

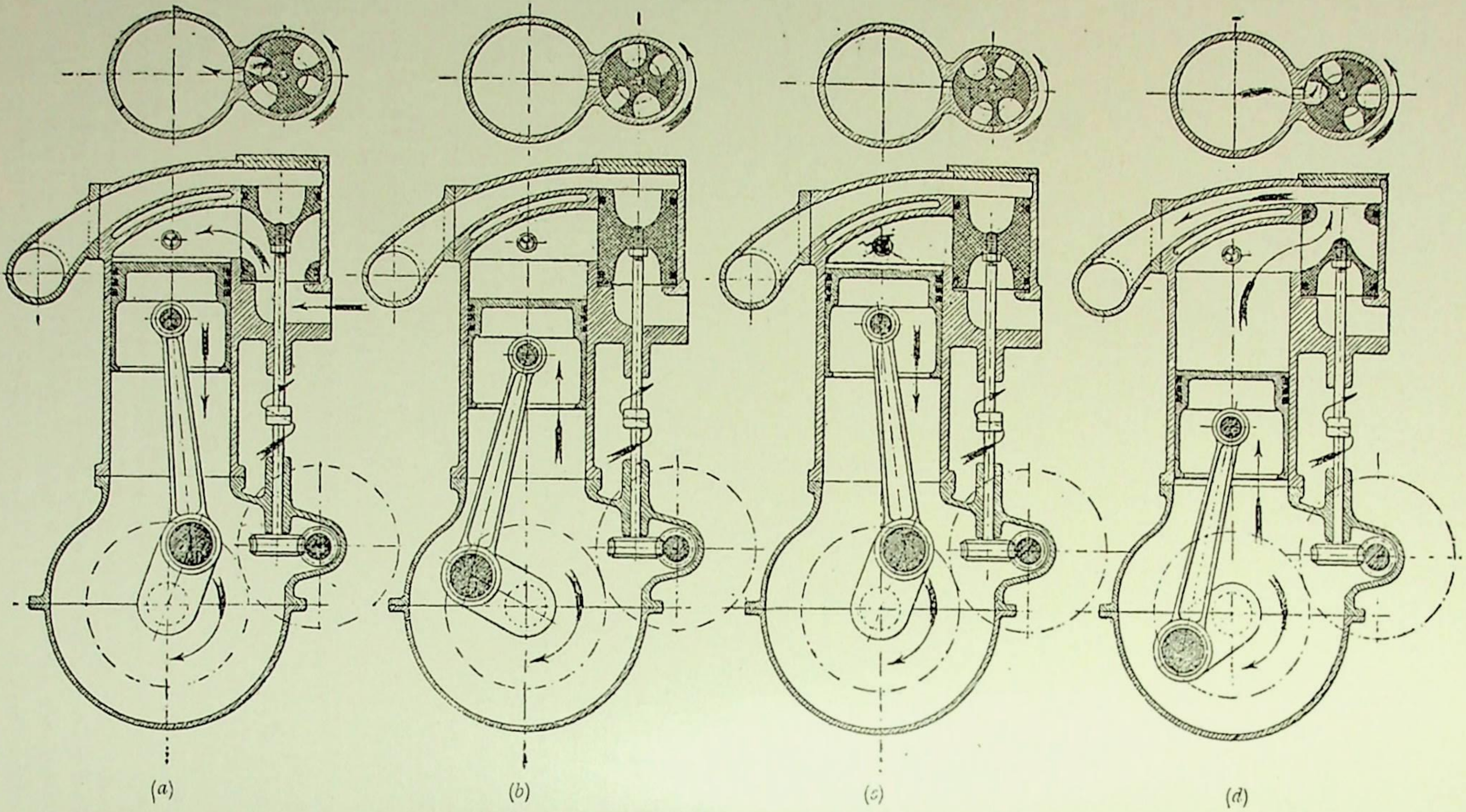
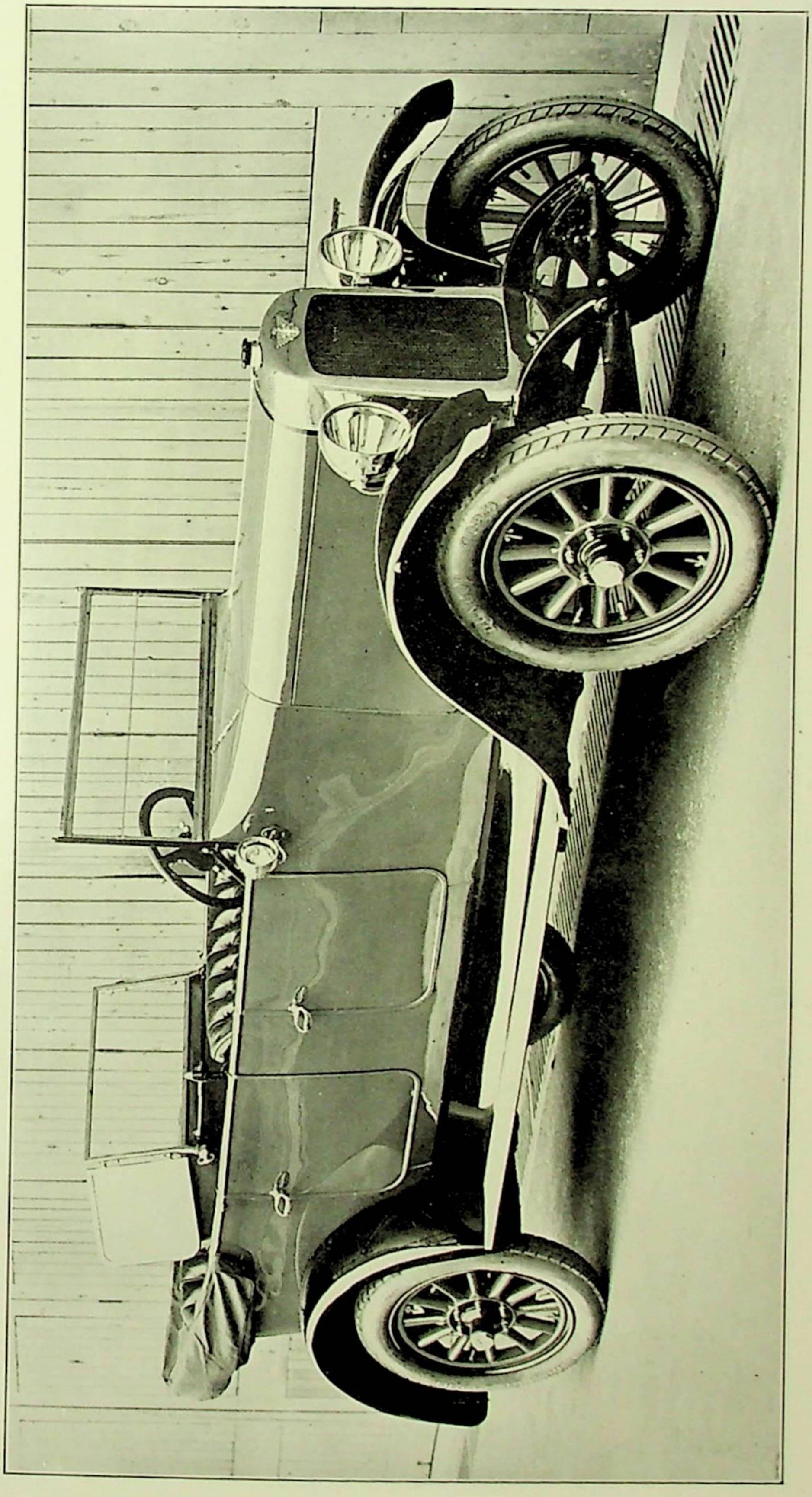


Fig. 142.—Itala Rotary Piston-valve Engine,



THE 20 H.P. AUSTIN TOURING CAR.

numbered 1, 2, 3, and 4 respectively. Each valve, with induction below and exhaust above, is fitted with two piston rings, one at either end, in order to prevent leakage, which might otherwise occur from the cylinder, along either path, during compression and firing operations. This use of piston rings for a rotating valve which has no longitudinal movement is rather uncommon, but nevertheless is quite effective, due to the fact that the rings do not rotate—the valve grooves slide over them. The combination thus becomes somewhat like a marine collar-thrust bearing and is very effective against end leakage. The valve body is also a good running fit in the casing, and means are provided in the design for obviating the excessive friction which would otherwise occur, due to side thrust under the influence of the cylinder pressure.

In the first place the valve is systematically lubricated by pressure-fed oil in a similar manner to the main bearings, so that it revolves without difficulty. In the second case an ingenious, yet simple, arrangement is provided for the purpose of counterbalancing the cylinder pressure on the valve during the firing stroke. A small hole is drilled through the valve body in such a position as to be open to the cylinder when explosion occurs; the explosion pressure is thus transmitted through this hole into the opposite side of the valve, and by recessing an area in the chamber wall equal to the cross-sectional area of the port, the resultant pressure on the valve is practically nothing. It will be seen that as no leakage takes place through the drilled hole or from the recess, this arrangement results in no loss of efficiency, and further, that owing to the small capacity of the recess, the loss of compression is very small. The shape of the combustion chamber in this engine is practically hemispherical. The valve itself was water-cooled.

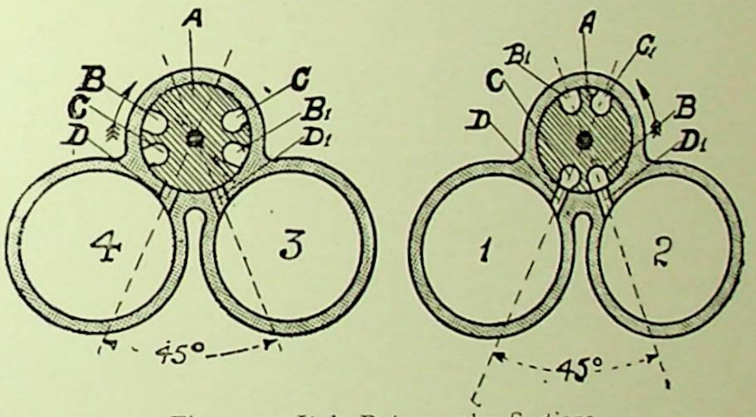


Fig. 143.—Itala Rotary-valve Sections.

The ordinary eight-cam shaft and valve-tappet gear are replaced in this engine by two high-efficiency worms driving the two rotary valves which control the operations of the four cylinders, so that an appreciable reduction in the complication of the ordinary poppet arrangement is obtained.

Referring to Fig. 142, the first diagram (*a*) on the left shows one cylinder with the piston at the top of its exhaust stroke, with the induction stroke about to take place, the valve port being open to the inlet manifold, and thence to the carburettor. Diagram (*b*) shows the piston coming up on its compression stroke, the valve being completely closed so as to allow the mixture to be compressed within the cylinder. In Diagram (*c*) ignition is just occurring and the piston commences to move down under the influence of the pressure of the exploding gases, the rotary valve still presenting its closed face to the cylinder, with the exception of the small hole, previously mentioned, for balancing purposes.

The exhaust operation is shown in Diagram (*d*), the valve port being in the appropriate position for communication between the cylinder and the exhaust manifold.

The engine in question was of the four-cylinder type, having a bore of 105 mm. and a stroke of 150 mm., and it enjoyed a period of popularity as a car engine.

Considering at this stage the principal difficulties associated with the use of rotary valves, these may be enumerated briefly as follows: (1) temperature difficulties; (2) defects due to lack of gas-tightness causing leakages of the gases between the cylinder, the exhaust, and the inlet ports; and (3) excessive wear due to the valve being subjected to explosive and compression forces.

The temperature differences of the exhaust gases and the fresh cool mixture are appreciable, and unless well designed the valve becomes distorted; moreover, the rotating valve may carry around with it exhaust particles and burnt oil, which result in undue wear taking place. The question of relieving the valve from the load of the explosion pressure is overcome in the Itala engine by the use of the drilled hole previously mentioned, and in the case of Henriod's engine by placing the parallel horizontal valve in communication (similar to that shown in Fig. 144) somewhat below the top of the cylinder, so that it does not have to sustain the full force of the explosion. The difficulties due to leakage, wear, and temperature action are not so important, and, moreover, have been satisfactorily overcome in the case of the poppet-valve engine, but at the expense of more complication in the mechanism and

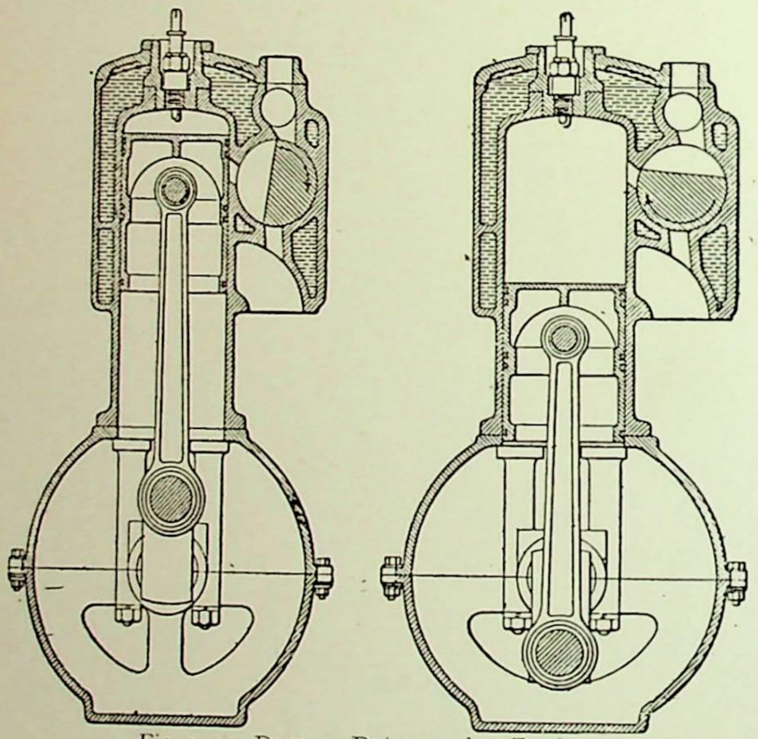


Fig. 144.—Darracq Rotary-valve Engine.

in the number of components, and in most cases of a somewhat noisier engine.

The rotary-valve engine, on account of its simplicity, is, as a rule, a cheaper engine to manufacture.

Fig. 144 gives two views, in section, of the well-known Darracq rotary-valve engine, the left-hand view corresponding to the end of the exhaust stroke, and the other view to the end of the suction and beginning of the compression stroke. In this case the valve is a long roller working in a cylindrical cavity with its axis horizontal. This rotary valve is driven at one-half engine speed by means of skew-gearing, and has ports cut in it in an appropriate manner for the various operations to occur.

It will be seen that the valve port to the cylinder is over-run by the piston, in order that the valve itself shall not be subjected to the full force of the explosion pressure, as in the Henriod case. It is obvious, however, that owing to this over-running of the port, on the exhaust stroke the port will be closed rather too early, and that there will be a larger quantity of exhaust gases trapped in the combustion head than is usually the case in poppet-valve engines. For the same reason the inlet operation cannot occur until the piston has descended a certain way, and a portion of the induction stroke will thus be lost. In such cases it is found necessary to employ rather larger cylinder dimensions for a given power than in the case of the poppet-valve engine.

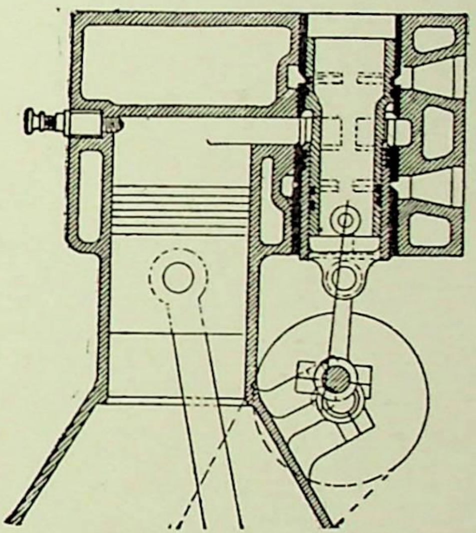


Fig. 144*a*.—An Example of a Cylindrical Slide-valve Engine.

CUFF-VALVE ENGINE

In the case of the *cuff-valve* engine a pair of concentric sleeves are operated in a co-axial chamber situated above the cylinder, and work in a somewhat similar manner to the double-sleeve valve engine. The sleeves are, of course, very much smaller than the cylinder, and are provided with ports for the admission and exhaust operations. The outer sleeve serves for the exhaust and the inner one for the inlet process. Both valves are raised when the exhaust operation occurs, but only the inlet-valve sleeve is moved when induction takes place.

SPECIAL TYPES OF FOUR-CYCLE ENGINES

It is now proposed to consider a few engines of special design, but working on the well-known Otto cycle. In most cases the ordinary poppet-valve system will apply, but it will be fairly obvious that any other valve arrangement could be employed, provided that the design difficulties were not over-riding.

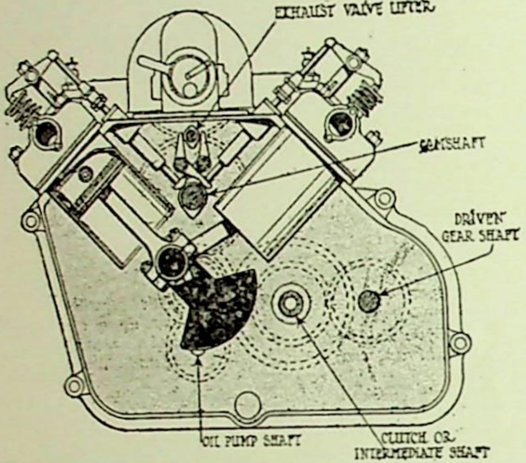


Fig. 145.—The Belsize-Bradshaw Oil-cooled Engine (*Autocar*).

The first engine of a somewhat unique design to be illustrated is the Belsize-Bradshaw one, which is used on the Belsize light cars. The special feature of this engine is that the lubricating oil in the crankcase is the principal cooling agency for the cylinders. The cylinders are so constructed that the greater portions of their long plain barrels protrude into the crank chamber, which is of fairly ample proportions, as will be evident from the illustrations. The lubricating oil is contained in a sump of about 1¼ gallons capacity, and is circulated at the rate of about one-half gallon per minute. A small-gear type of pump, situated on the timing-gear side of the engine, draws oil through an oil filter and delivers it under pressure to the hollow crankshaft, whence it passes to the big-end bearings and out again. By means of suitable channels or ducts arranged in the big ends and in the crankpins, three streams of oil are flung out on to the cylinder barrels and undersides of the pistons. These oil streams not only lubricate the sliding surfaces very effectively, but also convey the surplus heat away from these parts. The oil, which, of course, becomes warm as a result, is cooled again by contact with the walls of the crankcase, which are provided with longitudinal cooling ribs for the purpose.

The engine is of the Vee-twin type, with cylinders at 90°. The bore and stroke are 85 mm. and 121 mm. respectively, giving a cylinder capacity of 1,370 c.c., and a nominal horse-power of 9. The heads of the cylinders are the detachable type, in cast iron, and are provided with a parallel series of radiating fins. These cylinder heads contain the overhead valves and their operating rocker-arm bearings, and also the sparking plugs, so that the cylinder proper is of very simple shape, being merely a parallel machined cast-iron barrel with a square flange at the top, drilled and tapped with four holes for receiving the four holding-down bolts for the detachable heads.

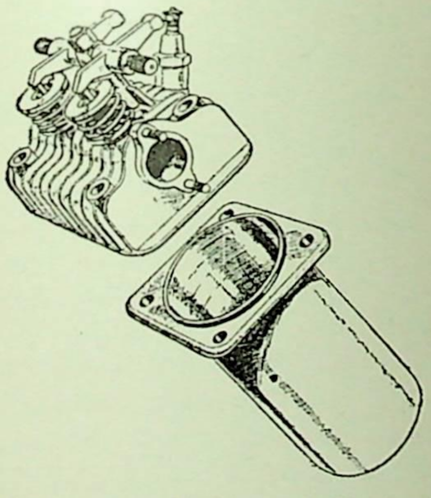


Fig. 146.—Cylinder and Head of Belsize-Bradshaw Engine (*Autocar*).

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The camshaft is parallel to the crankshaft, and situated just above it, and works the valve tappets (and push rods) through rockers. Above the main camshaft shown in Fig. 145 will be seen a secondary cam for lifting the exhaust valves by means of the extensions on the rockers. The pistons are made of an aluminium alloy, slightly concave at the top, short in length and provided with two rings above and one below the floating gudgeon pin, as well as a drilled oil groove at the base. Externally, this engine is somewhat unconventional in appearance, and it is only the short projecting cylinder heads which cause it to resemble a petrol engine.

Prolonged tests and experience show that the cooling system is very satisfactory, and that owing to the special design of the piston there is practically no carbonisation in the combustion head due to leakage of oil past the piston.

THE LANCIA STAGGERED VEE ENGINE

This engine is of the four-cylinder type, but instead of the cylinders being in a line as in the usual pattern, they are slightly inclined in pairs, and each cylinder is staggered relatively to its neighbour. It is made by the Lancia Company of Turin, and is fitted to their light car. Fig. 147 is a plan view of this engine, showing the top ends of the four inclined cylinders, whilst an outside view is shown in Fig. 148, the valve-gear covers being removed. The bore and stroke are 75 mm. and 120 mm. respectively, giving a cylinder capacity of 2,120 c.c. The nominal horsepower is 13·9, but the maximum exceeds 50 b.h.p.

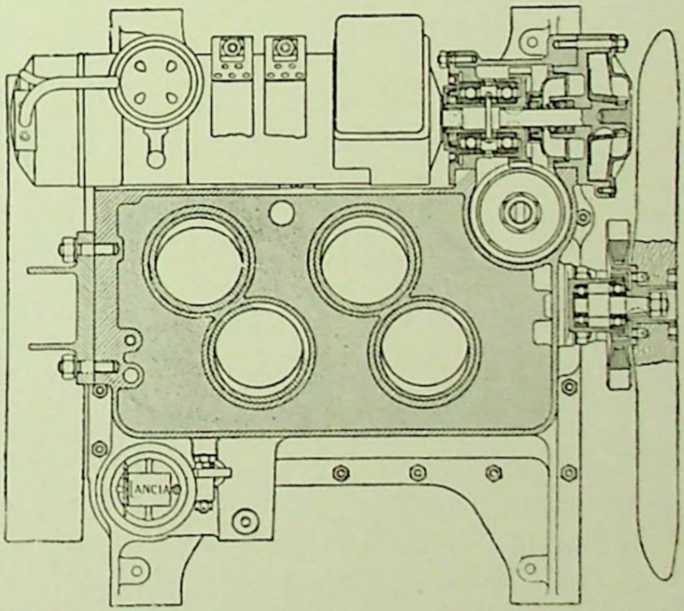


Fig. 147.—The Lancia Staggered Vee Engine, in Plan (*Autocar*).

The cylinder pairs are inclined at 14°, and owing to the stepped arrangement are all contained in one cylinder block of very short length, compared with the usual four-cylinder vertical type. The cylinder block has an overall length of only 16½ inches, the total length of the engine with fan and flywheel being 22 inches.

The cylinder barrels are of cast iron, and are made with a flange on the upper end of each barrel. They are roughly machined first, and are then set in a mould for the aluminium alloy (for the surrounding water jackets, etc.) to be cast around them. In this manner a combined aluminium-iron cylinder block is obtained with most of the advantages of both. There is the usual type of copper and asbestos joint between the cylinder block and the detachable head, but the cast-iron cylinder barrel and the cast-iron head are in contact. The usual type of forked connecting rods are not used on this engine; for, owing to the staggering of the cylinders, it is possible to employ rods of the orthodox pattern with a separate crankpin for each. The aluminium pistons are die cast and provided with three steel piston rings of 2 mm. depth only, and in addition there are fitted two special scraper rings purely for oil-reduction purposes.

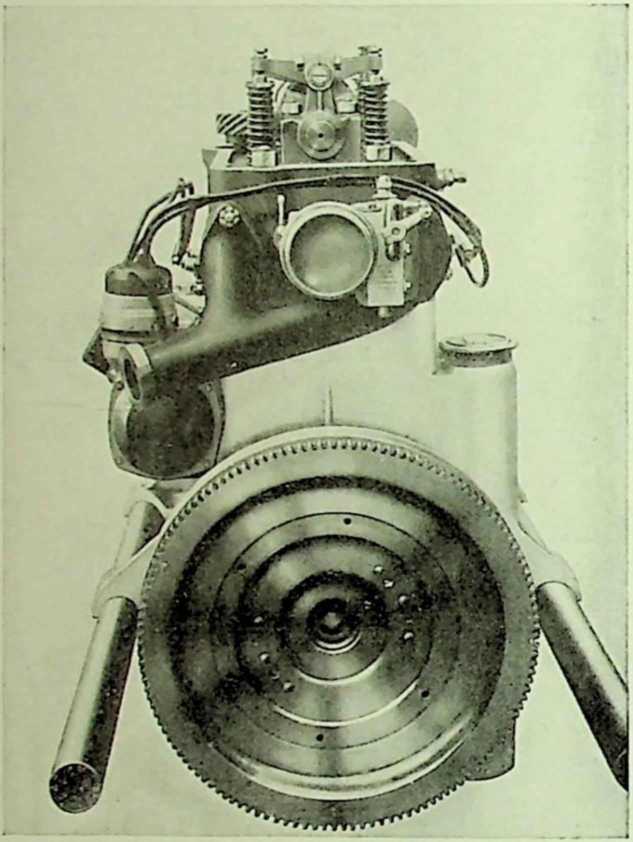


Fig. 148.—The Lancia Engine.

Overhead valves, operated by an overhead camshaft, are employed, and double valve springs are fitted to each valve, so that spring periods are avoided, and in the event of one spring breaking, the other one can carry on satisfactorily for a time.

THE RADIAL-TYPE ENGINE

Although this design of engine is still widely employed upon aircraft, and has appeared from time to time on light cars, yet it has not had the vogue which its merits would appear to warrant.

The simplest type of this engine is the two-cylinder opposed engine, although, strictly speaking, there should only be one crankpin instead of the two at 180° generally used, in order to come within this category.

The radial engines usually adopted comprise the three, five, seven, nine, etc., cylinder star-types, with one master connecting rod and a series of auxiliary rods hinged to the master rod, all working upon one crankpin. A very compact engine of approximately circular overall form is thus obtained, and one which is fairly simple to construct, more particularly in regard to the cylinders (which can be made interchangeable) and to the valve cams (which can be formed on a single cam-wheel).

The types which have been employed upon light cars were air cooled and of the three- and five-cylinder types; it is necessary to have an odd number of cylinders in order to obtain even firing intervals. Thus, in the case of the three-cylinder radial engine the cylinders are set at 120°, and the firing intervals are 240°; with the five-cylinder type the angles between the cylinders are 72°, and the firing intervals 144°, or the order of firing, if the cylinders are numbered from 1 to 5, in a clockwise direction is 1, 3, 5, 2, 4, there being five explosions every two revolutions.

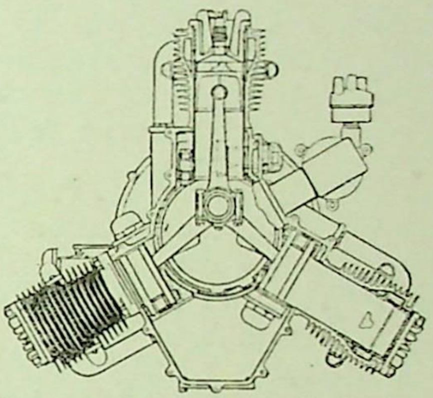


Fig. 149.—Three-cylinder Single-sleeve Valve Engine.

The general arrangements of three- and five-cylinder engines of the radial type are shown in Figs. 149 and 150.

The early difficulties with this engine were due to lubrication troubles, it being not easy to ensure an even supply of oil to each of the cylinders, more particularly the lower ones. The exhaust piping presents a little complication, and this type of engine is usually somewhat noisy. The balance of the radial engine with an unequal number of cylinders is quite good. Thus in the case of the seven-cylinder type the reciprocating forces are in balance, and also the secondary ones.

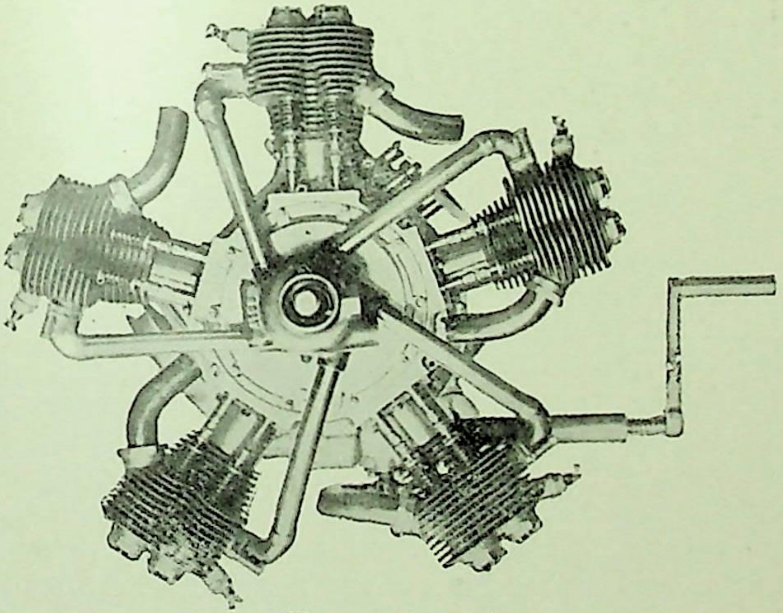


Fig. 150 (*Autocar*).

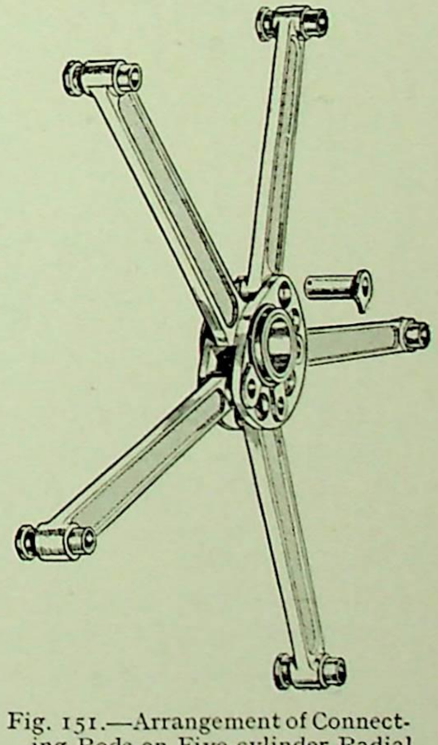
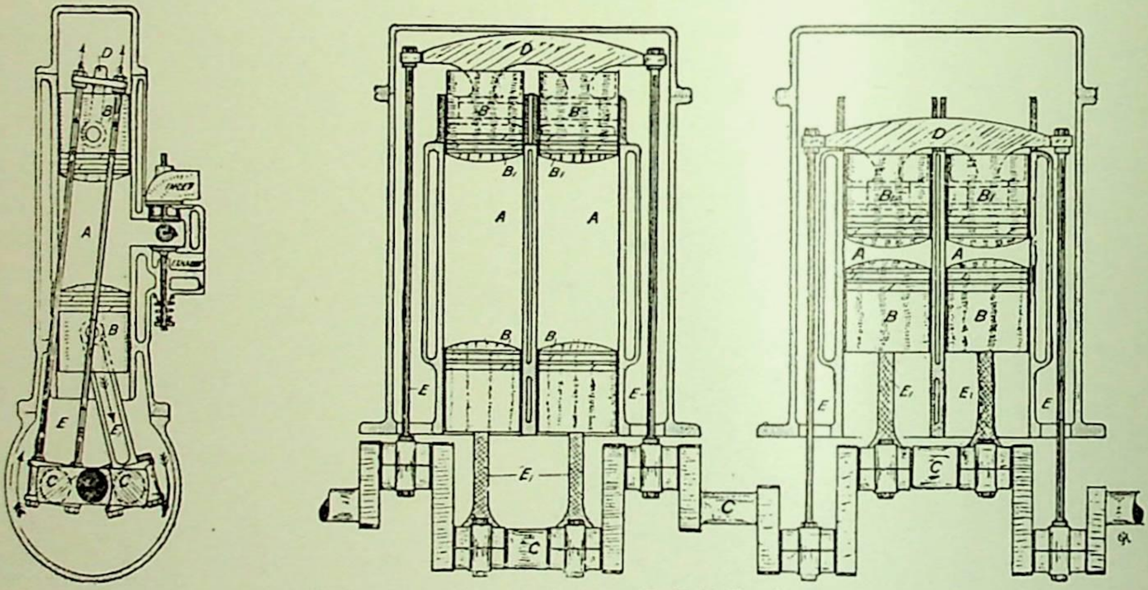


Fig. 151.—Arrangement of Connecting Rods on Five-cylinder Radial Engine (*Autocar*).

The arrangement of the connecting rods, to enable them to work on the one crankpin, is of interest. There is one main connecting rod, as we have stated, from one of the cylinders, which is a bearing fit on the crankpin. This rod is provided at its big end with a number of bosses, each with a hole in to receive the pin of the auxiliary connecting rods. There is thus only one rod bearing directly upon the crankpin, whilst all the other rods are hinged to it and revolve upon their own pins.

THE DOUBLE-PISTON ENGINE

An interesting engine, which was employed for a fairly long period upon a French automobile, namely the Gobron-Brillié, possesses certain unique features of both theoretical and practical interest, and its description will serve as an excellent example of one of the few means of obtaining almost perfect engine balance. The four-cylinder type of Gobron engine is illustrated in Figs. 152 and 153 in side and front sectional elevations, respectively.



Figs. 152 and 153.—Gobron-Brillié Engine.

Each cylinder is unusually long and accommodates two pistons, which reciprocate in opposite directions, but symmetrically, about a central position. The lower pistons, B1, are attached to the crankshaft in the ordinary way by means of connecting rods, E1, while the upper pistons B are coupled together in pairs by means of the cross-pieces or rigid arms D, which are attached to the crankshaft (but on different crankarms at 180° to those of the lower pistons) by means of the skeleton rods marked E. The combustion chamber is the space A between the heads of the opposing pistons. It will be obvious that each pair of pistons approach or recede as the engine works, and that if the weights of the reciprocating masses are equal, the reciprocating forces will be completely balanced. Further, since each piston in a cylinder is of the same area, and is under the same explosion pressure, the forces transmitted to the crankshaft, due to the explosion, will be practically identical, while the rocking actions due to their being out of line also balance, so that no hammering action is felt on the crankshaft. The reciprocating forces, in the strictest sense of the word, are not completely in balance, due to the different ratios of the connecting rods of the opposing pistons to their cranks, but the difference due to the angularities of these rods is not serious.

Summing up the advantages of this type of engine, they are as follows: (1) Almost perfect balance of the reciprocating parts. (2) Balance of the explosion and compression-pressure effects on the crankshaft. (3) Better expansion due to the long stroke-bore ratio. The disadvantages which might be cited against this type of engine are its overall height, which necessitates high bonnets, and its rather greater weight, due to the construction.

THE DESAXÉ ENGINE

Although this is a modified design, or type, of the ordinary engine, it has been included here rather than in an earlier section on account of its differing from the standard type in the matter of the cylinder position.

This offset or *desaxé* type of engine has its cylinders set forward, parallel to the usual symmetrical position, and in the direction of rotation, by a small amount as shown in Fig. 154. This offset is usually from $\frac{1}{5}$ to $\frac{1}{6}$ of the cylinder diameter.

The principal object in adopting this arrangement is to reduce the obliquity of the connecting rod during the firing stroke, and hence to reduce the amount of piston thrust on the cylinder walls. That this is the case will be evident from a consideration of the line diagram given in Fig. 77.

Another advantage of this type is that the motion of the piston is slower at the commencement of the firing stroke, so that a rather higher explosion pressure will result.

The stroke of the piston of an offset engine is a little longer than that of the normal type, with the same connecting-rod length and crank throw; thus in the case of an engine with a ratio of connecting rod to crank of 4, the offset piston stroke is about 0·3 per cent. greater. It will be evident that the obliquity of the connecting rod on the compression and exhaust strokes will be greater than in the case of the normal engine, but as the compression and exhaust pressures are so very much smaller than the explosion pressure,

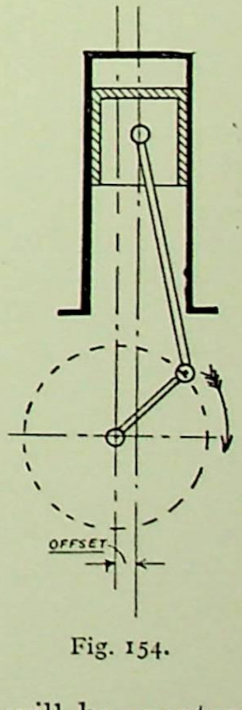


Fig. 154.

the greater thrust due to the former pressures will be very considerably outweighed by the reduced thrust on the explosion stroke.

The Chapuis Dornier, and other motor-car engines, together with certain aircraft engines, have been made on the offset principle with beneficial results.

OTHER TYPES OF PETROL ENGINE

It would be impossible to deal with each and every type of engine which has been suggested, patented, or even made, in the space at one's disposal here, and for this reason only those which have shown commercial promise or success, or which possess unique features illustrating the application of principles which it is believed will be helpful to the reader to follow, have been described in the present chapter.

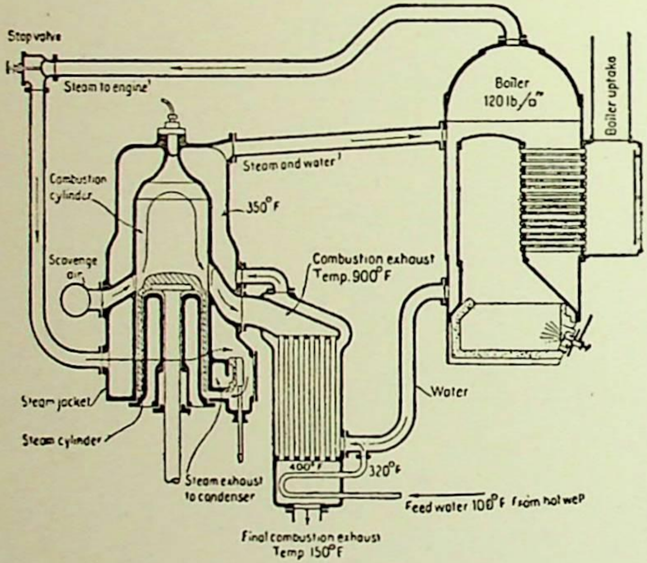


Fig. 155.—Showing Principle of the Still Engine.

Numerous attempts have been made to improve the efficiency of the ordinary engine by endeavouring to utilise part of the waste heat of the gases and the cooling water. Most of these attempts have failed for one reason or another, connected with mechanical complications, and loss of the "gain" in driving the extra parts. There is one engine, however, which has met with a considerable degree of success in this direction, namely, the Still engine. Although it has not yet been applied to automobile purposes, it has proved very satisfactory in land and marine power-plant work

Briefly, the principle of the engine is as follows: The water jacket is in direct communication with the boiler of a steam engine, so that the jacket water is also the boiler water and the steam. The heat of the cooling water, which is derived from the energy of exploding and expanding gases, is, therefore, recuperated. Further, the exhaust gases are employed to generate steam in the boiler, and afterwards to heat the feed water to the jacket-boiler combination. The steam thus generated acts upon the lower side of the piston in the petrol or other type of internal-combustion engine employed, and therefore does additional useful work on it.

As the exhaust gases are at a temperature of from 700° C. to 900° C., it will be evident that there is ample heat to obtain steam from the boiler. Moreover, it should be remembered that for every useful horse-power obtained in a petrol engine no less than three horse-power is wasted in the heat carried off to the jacket water (and the surrounding air), and by the escaping exhaust gases. With the Still engine it has been shown that from 15 to 30 per cent. more power can be obtained for the same petrol or fuel-oil consumption.

Thus in the case of a 5-inch bore by 5-inch stroke Still petrol engine, the average pressure from the exploding gases over the stroke was 90 lb. per sq. inch, whilst on the steam side of the piston a further average pressure of 14 lb. per sq. inch was obtained, and the net gain in power over the petrol engine, due to the double stroke, was nearly 30 per cent. By burning fuel in the boiler additional steam could be generated and the steam pressure increased to 72 lb. per sq. inch, which was equivalent to a total increase in power of about 100 per cent. as compared with the normal petrol engine.

When the Still principle is applied to the Diesel engine, very high efficiencies—in fact, the highest yet obtained by any means—result: in one or two cases thermal efficiencies of about 40 to 45 per cent. have been measured, as compared with the 20 to 27 per cent. obtained in the case of the petrol engine, and 15 to 18 per cent. for the most efficient steam engine (i.e. the steam turbine).

SUPERCHARGED ENGINES

Hitherto the question of obtaining extra power from the petrol engine, by means other than those usually employed (for example, by increasing the compression and speed, etc.), and without the aid of external means, has been chiefly of importance in the aeronautical world, but recently engines have been made and fitted to automobiles which work on what is known as the supercharged principle. Briefly speaking, supercharging consists in supplying to the engine carburettor system air at a higher pressure than the outside atmospheric pressure, the power necessary for compressing the air being recuperated from the exhaust, as a rule.

Tests which have been made upon aircraft engines show that an increase in power output of from 20 to 30 per cent. can be obtained by this means. In some cases very high-speed centrifugal blowers are employed, and are gear-driven from the engine itself; in other cases the exhaust gases are led into a Rateau type of turbine which revolves at a speed of from 15,000 to 30,000 r.p.m., and the centrifugal blower is coupled direct to the turbine rotor shaft.

In the case of the Mercédès 10–40 h.p. car a supercharging device, which has been standardised, consists of a gear-driven air blower, which supplies air under pressure to a special type of carburettor, closed to the external atmosphere when supercharging occurs. An additional warm-air device, exhaust heated, is embodied in the system.

Fig. 156 illustrates the principle of this device, and shows the hot and cold air inlets, and on the near side the large suction pipe, which is taken to the blower. The delivery pipe from the blower to the carburettor is on the top right-hand side. The normal air-inlet pipe runs between the cylinders direct to the carburettor and has a butterfly type of throttle valve fitted inside.

The supercharger itself is driven off the front end of the engine, being mounted on the crankcase. The drive for the blower consists of a multiplate clutch and bevel gearing, and the blower is contained in a small aluminium casing mounted on top of the drive casing with its spindle vertical.

Although a supercharged automobile engine is undoubtedly superior from the point of view of power output for a given capacity, and would be a distinct advantage for racing or taxation-evasion purposes, yet it is very improbable that it will come into general use, for a small increase in the dimensions of an engine will meet any demand for extra power without the complications of the supercharged engine.

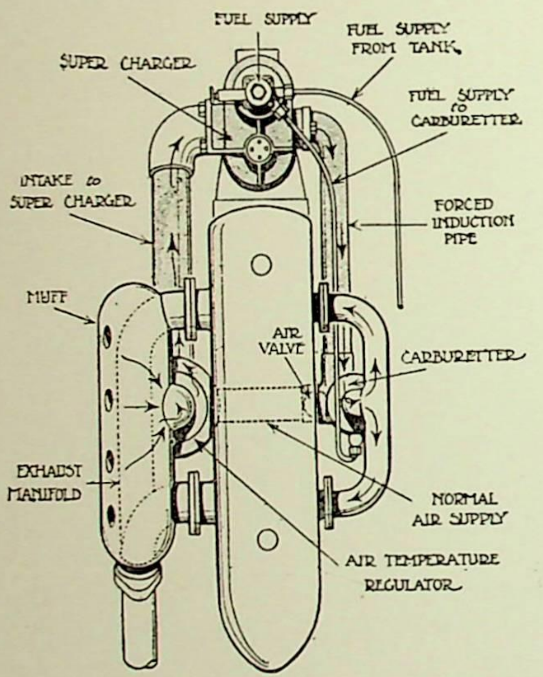


Fig. 156.—The Mercédès Car Engine Superchanger (*Autocar*).

Another interesting example of a supercharged engine was the Ricardo. This engine was of the four-cycle type.

The cylinder proper, at the upper end, was provided with a lower extension, consisting of an inner barrel which served as a guide to the trunk of the piston, and an outer barrel or casting, in the annular chamber of which air was compressed to about 12 lb. per sq. inch. The principle was as follows:

The piston in descending on the suction stroke drew in the mixture from the carburettor in the usual manner through the inlet valve in the cylinder head, and the annular piston at the same time compressed pure air in the space between the lower inner barrel and outer cylinder. Just before the main piston uncovered the ports leading from the compression chamber to the main cylinder, the inlet valve was closed, so that when the ports were uncovered, compressed air rushed into the cylinder and increased the inlet charge pressure to about 5 lb. per sq. inch.

On the exhaust stroke the air was again compressed, and on the ports being uncovered, the compressed air scavenged the cylinder of burnt gases, sweeping most of these out through the exhaust valve. The usual method employed was to allow a rich mixture to enter from the carburettor, so that the dilution due to the entering compressed air was just sufficient to give normal proportions. The results of tests made by Ricardo on this engine showed that the maximum b.h.p. at 1,000 r.p.m. was increased from 11·2 to 14·7, and at 2,200 r.p.m. from 20·5 to 30·5—an increase of 50 per cent. The fuel consumptions at speeds of 2,000 r.p.m. were 0·535 and 0·470 lb. per b.h.p. hour respectively, thus showing a marked economy in the case of the supercharged engine. It will thus be seen that the additional power was not obtained at the expense of a sacrifice in thermal efficiency.

The dimensions of the single-cylinder engine tested were 110 mm. bore and 150 mm. stroke, the normal compression ratio being 5.

AIR-COOLED AUTOMOBILE ENGINES

Reference has been made already, upon several occasions, to the design features and arrangement of air-cooled engines, from the point of view of motor-cycle and light-car use. There is another field in which this type of engine is beginning to obtain a firm footing, namely, in the case of the ordinary motor car.

For several years in America the Franklin air-cooled engine has been in use upon a high-powered and expensive car, with every success, and recently the Chevrolet car [1] has adopted air cooling. The pre-war Cameron air-cooled car engine, with its four vertical cylinders, and its more recent development a six-cylinder vertical air-cooled engine of high power, are other examples. In the light-car world, also, the air-cooled engine has proved itself to be a serious rival of the water-cooled type in this country, the types most favoured being the horizontally opposed and the Vee-twin models. It is chiefly in America that the larger-powered air-cooled cars are in use, and they appear to have reached a fairly high standard of development in that country. For motor-cycle use the four-cylinder engine has already proved satisfactory.

The early Franklin air-cooled engine system had a series of vertical cowls or sleeves surrounding the cylinder, through the annular spaces between which the cooling air is drawn by means of a suction fan situated to the rear of the engine (i.e. on the dashboard side). The cylinders are of cast iron with steel cooling fins placed radially and axially ; these fins are placed in the mould prior to casting. The inset diagram on the top right-hand side shows a section through the cylinder, with the cooling fins and the surrounding cooling cowl in place. The latest Franklin six-cylinder engine dispenses with the suction fan and employs a positive pressure blower on the forward end of the engine, which forces air down the cowls, and out past the lower side of the crankcase. This system was adopted in view of the fact that the flywheel suction fan could not be made of sufficiently large diameter owing to space restrictions.

The engine is of the six-cylinder vertical type of 82·5 mm. (3¼ inches) bore by 101·6 mm. (4 inch) stroke, giving a cubic capacity of 3,260 c.c. (199·1 cu. inches). The fins are radial, as previously mentioned, and each steel fin is turned over at right angles at its outer edge, so as to touch the next fin and thus to form an enclosed cylindrical space between the outer edges and the cylinder. The length of each fin down the cylinder is about 5 inches and its width about 1 inch. The engine is fitted with overhead valves operated by push rods. A complete description of this engine will be found in *The Automobile Engineer* for February 1923.

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[1] American model.

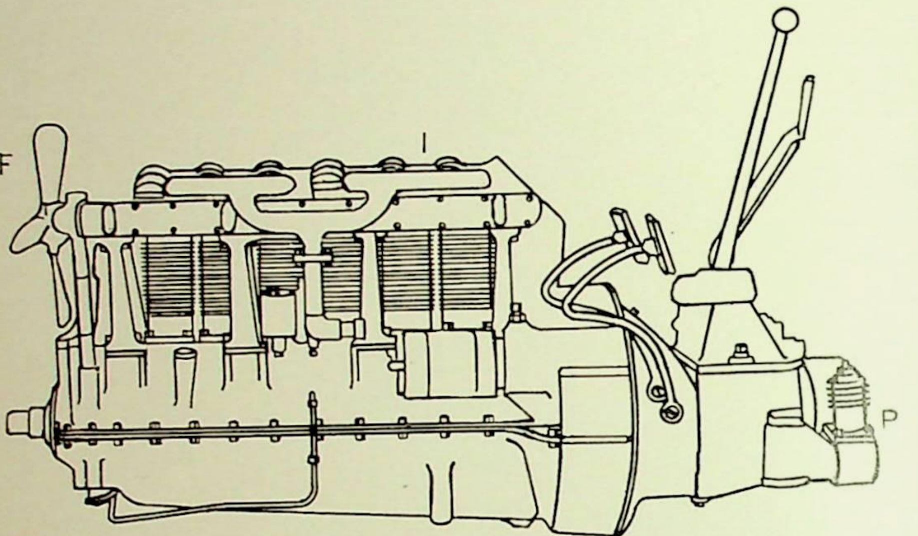

Fig. 157.—The L.W.H. Six-cylinder Air-cooled Engine.

Fig. 157 is an outline illustration of the L.W.H. six-cylinder air-cooled motor, which is direct-cooled by means of air which is drawn in from the front end by the fan, F, shown on the left and is ejected through the fly-wheel. The diagram shows also the induction-pipe system, I, with its two main branch pipes, each supplying three cylinders. The valves in this engine are placed horizontally in pockets formed in the head of the cylinder, and they are operated by means of a horizontal camshaft on the same level. The exhaust pipe which runs along the top of the cylinder is provided with cooling ribs. In the diagram the clutch and brake pedals, and the central gear control and hand-brake levers, can be seen. To the rear of the integral gear box is an air pump, P, for supplying air to the tyres, and fuel tank.

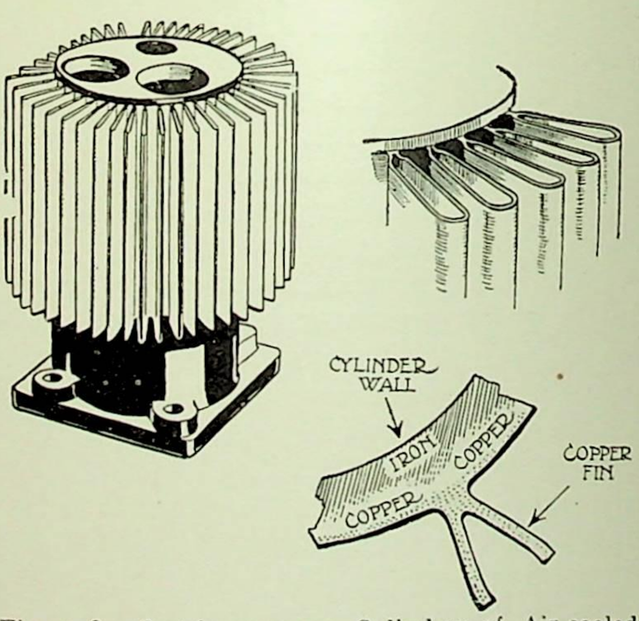

Fig. 158.—Cast-iron-copper Cylinders of Air-cooled Chevrolet Engine (*Autocar*).

In the case of the four-cylinder air-cooled Chevrolet engine, the cast-iron cylinders are provided with copper cooling fins, welded in place, and the latter are disposed around the cylinder barrel in the manner shown in Fig. 158, which diagram shows also one of the cylinders with the two overhead valve openings.

For the cooling system the air is drawn through the radiator (or the louvres in the dummy radiator) and passes axially upwards between the cylinder outer cowling and the cylinder walls, past the copper fins, and along the tops of the cylinders, out through the suction fan, which is in the front of the engine.

It should be pointed out that the conductivity of copper is considerably superior to iron, and is about twice as good as aluminium, so that it is an excellent material for cooling fins. The relative conductivities of copper, aluminium, and iron are in the approximate ratio of 100 to 50, to 22, so that the reason for using the first metal in the present example will be evident.

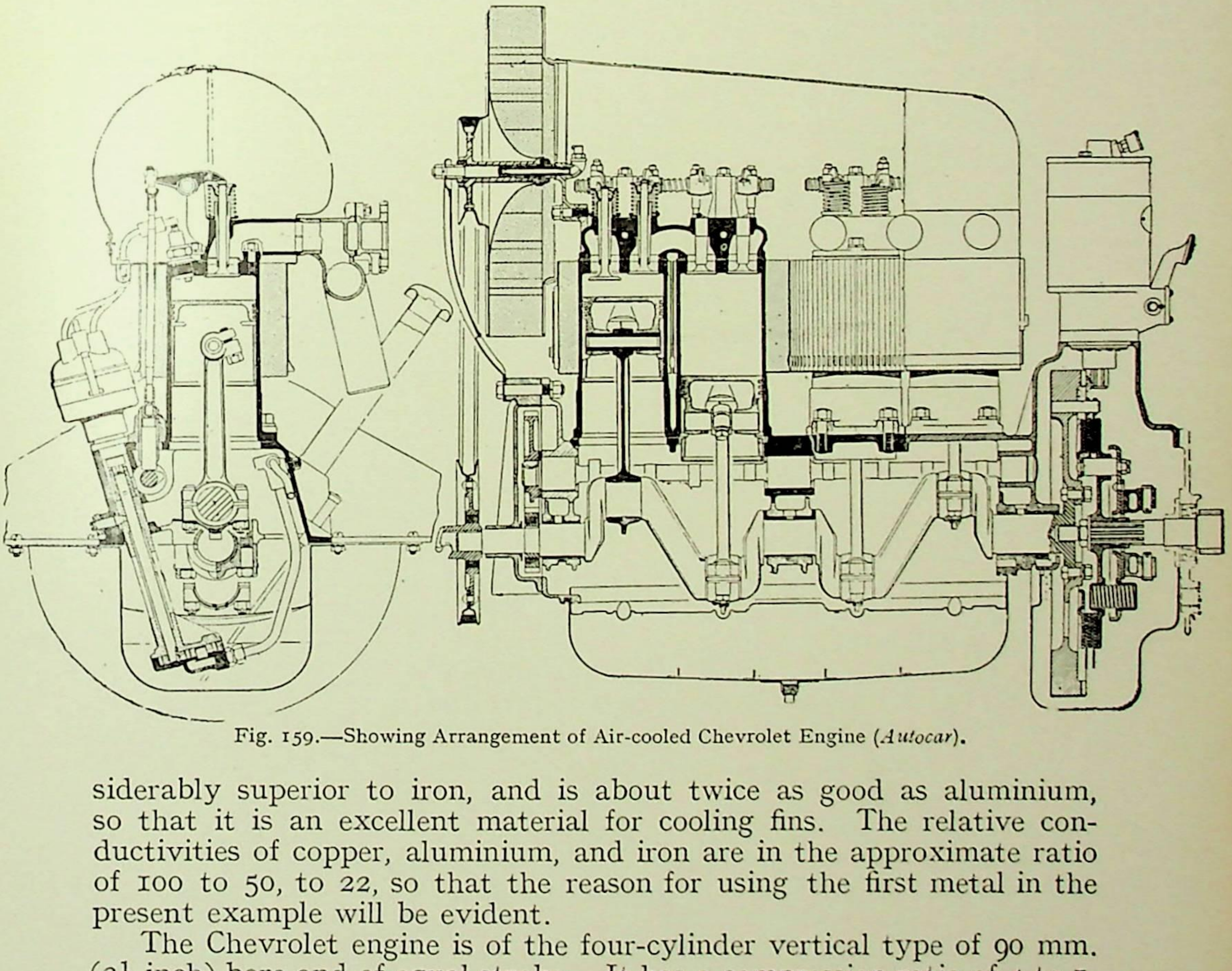

Fig. 159.—Showing Arrangement of Air-cooled Chevrolet Engine (*Autocar*).

The Chevrolet engine is of the four-cylinder vertical type of 90 mm. ($3\frac{1}{2}$ inch) bore and of equal stroke. It has a compression ratio of 4 to 1, and gives about 22 b.h.p. at a speed of 1,750 r.p.m. The car to which this engine is fitted has a special induction system, so designed that it prevents the full power of the engine from being realised, the maximum road speed being 45 m.p.h., whereas the engine with unrestricted induction will attain a speed of 60 m.p.h.

An interesting feature of this engine is the adoption of a form of *oil-less* bearing for the valve rockers, on account of the difficulties of supplying lubricating oil to the heads of the cylinder. The bearings are made up of a metal which consists of a combination of powdered graphite and very finely divided bronze which are mixed together, moulded to the shape required, and heated to the fusion point of the bronze. The Sirocco-type blower is driven at one-and-one-third engine speed by means of a Vee rubber belt of the motor-cycle type.

Generally speaking, the air-cooled motor-car engine has proved itself satisfactory in service, although in numbers and in types it is still a very long way behind its water-cooled rivals. That it will work continuously under heavy load if properly designed is shown by the results of tests made with the Franklin car in 1919. This car, which was a standard-production model, was driven through Death Valley, in America, with an atmospheric temperature of 120° Fahr., and up Pike's Peak to 14,900 feet elevation on a low gear without any overheating or other troubles.

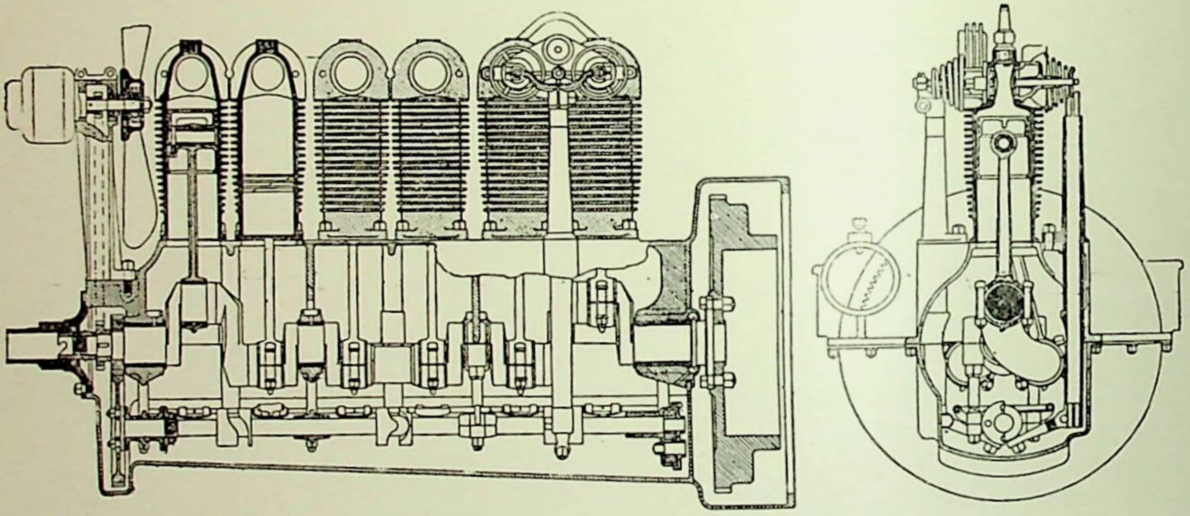

Fig. 160.—The Cameron Six-cylinder Engine (*Autocar*).

The air-cooled engine is cheaper to manufacture than the same size of water-cooled engine, if the preliminary design is properly arranged, and it has the advantage that it warms up more quickly from the cold, and also on account of its being able to withstand temperatures below freezing point without the cylinders cracking, as would be the case with water-cooled engines in which the water was inadvertently left in the jackets. The air-cooled engine requires a greater effort to start in cold weather on account of the thicker lubricating oil, but it works at a rather higher cylinder temperature as a rule. For a given capacity the water-cooled car engine is superior in the matter of power output, for higher compressions can be employed, and there is less tendency to pre-ignition and to loss of charge weight due to premature heating (and expansion) of the mixture.

Generally speaking, the air-cooled engine, owing chiefly to its reduced mass of " damping " metal, is more noisy than the water-cooled type.

Alternative methods of constructing the air-cooled cylinders to those previously described are by the use of steel cylinders machined from a solid bar or forging, so as to obtain a thin but strong section, and by the use of aluminium castings for the cylinders and heads, but with the insertion of steel or cast-iron liners for the cylinder barrels.

THE TWO-STROKE ENGINE

The principle of this engine has already been referred to, and the different designs of two-stroke engine which have been, and are now being, employed differ only in the means which are used for introducing the mixture into the cylinder, in the timing operations, and scavenging methods employed.

Unfortunately, this type of engine has hitherto only been adopted where reasons of cheapness have ruled, and it has not been developed to the same degree as the four-cycle engine. The difficulties in obtaining high mean effective pressures are very serious, and many able brains have attempted, with varying degrees of success, to improve the two-stroke engine for automobile purposes. It is possible, therefore, that in the near future this type will receive greater attention from the point of view of improving its performance.

The difficulties which have been found in the case of the two-cycle engine are as follows: (1) Low mean effective pressure, due to bad charging of the cylinder and to bad scavenging of the exhaust gases. (2) Cooling difficulties due to the fact that between 50 and 70 per cent. more heat is developed and must be conducted away in the same time as in the case of the four-stroke engine. (3) Want of flexibility; that is to say, inability to run regularly and smoothly over a wide range of speeds, and in particular to fire evenly at slow speeds or light loads. (4) Greater fuel consumption than in the case of the four-stroke engine.

Against these not-insuperable difficulties may be set the following advantages: (1) Higher power output for a given cylinder capacity. (2) Lighter engine for a given power output. (3) More even torque due to double the number of firing impulses. (4) Less expensive engine, generally speaking, to manufacture.

Unfortunately, there are few two-stroke engines employed upon modern light cars, but there are still cars, such as the " Valveless " type in use, which employ this engine type. Amongst modern examples in this country are the " Trojan " light car engine, and the Scott Sociable engine; whilst in France, to a two-stroke engined car belongs the distinction given of winning one of the big road races a year or two ago.

Figs. 161 and 162 illustrate in cross-section the " Valveless " engine made by Messrs. David Brown, of Huddersfield.

This engine, it will be seen, consists of two crankshafts geared together, so that they rotate in opposite directions. There is a common combustion chamber at the top, and crankcase compression is employed. Referring to Fig. 161, which shows the two pistons descending on the firing stroke and one uncovering the exhaust ports in the cylinder walls, it will be noticed that the other piston has uncovered the inlet port, and that the slightly compressed charge of petrol and air is entering the cylinder, displacing the remaining burnt gases through the exhaust ports E. The pistons then reascend, closing the ports, compressing the charge, and drawing in a fresh charge through the air valve shown in Fig. 162 past the disc D and through the passage S into the crankcase. The needle N being lifted by the disc, actuated by the suction, a jet of petrol is thus sprayed into the throttle passage P. Air alone is drawn into the crankcase and compressed ; the petrol remains in the passage P until the air from the crankcase sweeps it into the cylinder on the inlet port opening at T.

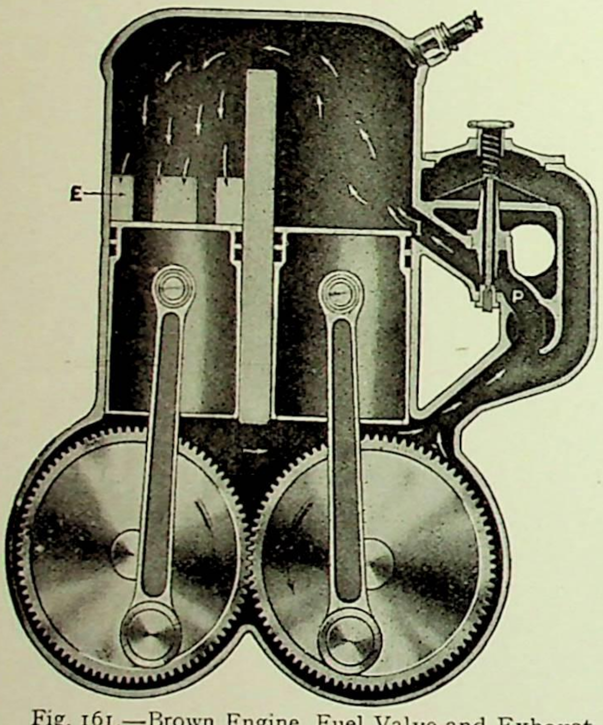

Fig. 161.—Brown Engine, Fuel Valve and Exhaust.

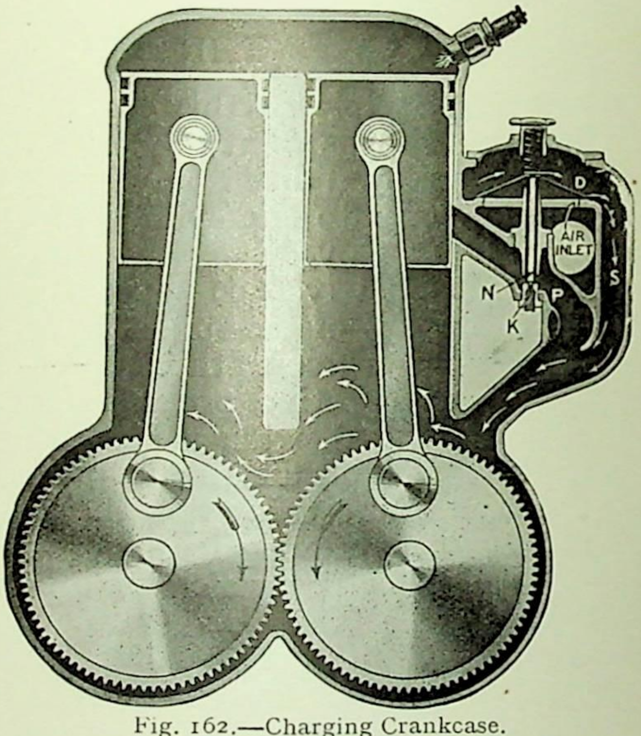

Fig. 162.—Charging Crankcase.

A charge is compressed and fired in the cylinder while the next charge is being prepared in the crankcase. The first charge at starting is obtained by turning the starting handle, whilst air under pressure is supplied to the petrol tank from the crankcase.

Owing to the system of geared crank discs employed, it is possible to balance the reciprocating forces, by the addition of a pair of balance weights, one on each of the two discs, and symmetrically disposed to one another. Each balance weight will then be on the opposite side of the crank journal to the crankpin, and the common centre of gravity will reciprocate up and down but always in a direction opposite to the pistons.

Although this type of engine is an improvement on the one previously described, yet it is not free from the objection of charge-loss due to the inlet and exhaust ports both being open at the same time. The arrangement of the $3\frac{3}{4}$ h.p. Scott two-cylinder two-stroke motor-cycle engine is shown in Figs. 163 and 164. This engine, which is of the three-port type, has been highly satisfactory both in competitions and in general service.

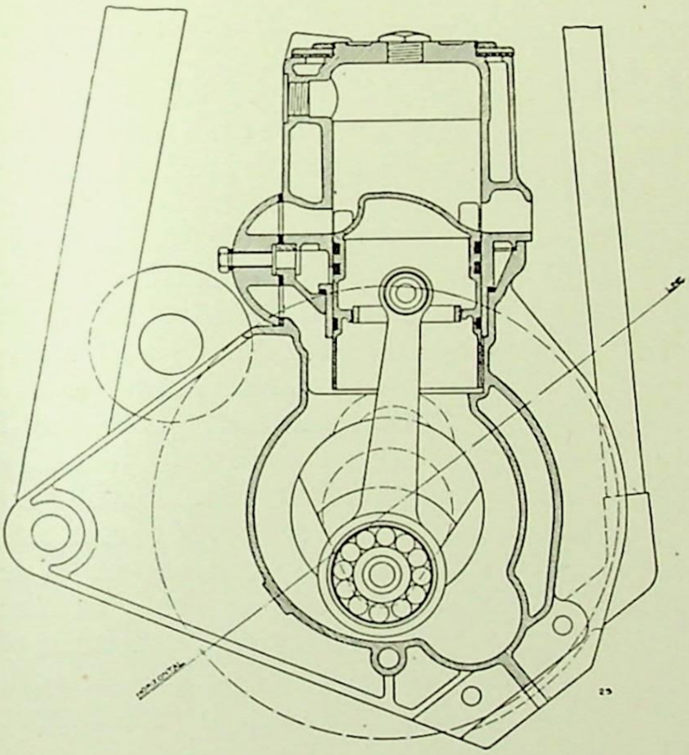

Fig. 163.—The Scott Two-stroke Two-cylinder Engine.

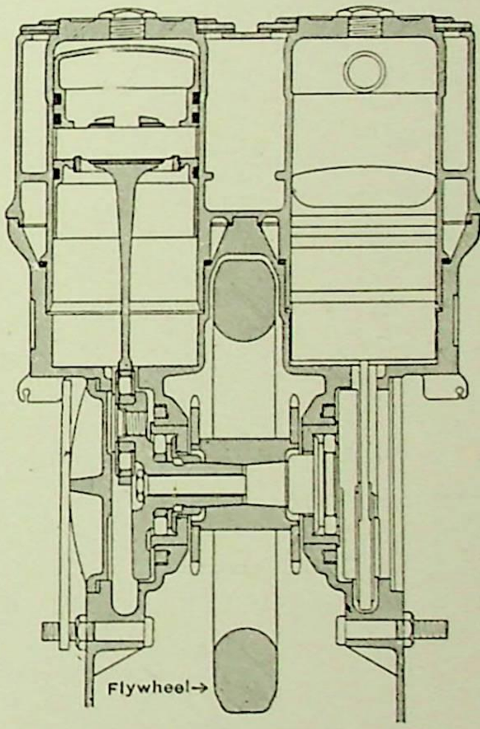

Fig. 164.—The Scott Two-stroke Two-cylinder Engine.

THE LEYLAND TWO-STROKE ENGINE

An interesting and very satisfactory two-stroke engine is that fitted by the Leyland Motor Co. to their "Trojan" light car. The engine is of the four-cylinder horizontal type, the bore and stroke being 64 mm. and 121 mm. respectively. The cylinders are water cooled, and are arranged in pairs side by side. Each pair of cylinders in reality act as a single cylinder and each pair fire alternately. The object of adopting the double-cylinder idea is to obtain better scavenging, and is somewhat similar to the "Valveless" arrangement.

Crankcase compression is employed, each pair of cylinders having a separate crankcase. The cylinders of a pair are vertically one over the other, and the cranks of each pair are at 180° to each other. The mixture of petrol and air is drawn into the crankcase while the pistons are moving outwards, and pumped into the upper cylinder (of the pair) as the pistons descend. It enters the second cylinder through an orifice in the combustion head.

The lower piston, which governs the exhaust port, moves in advance of the upper piston (governing the transfer port), so that exhaust occurs before the transfer port opens. As the ports are in the opposite cylinders of a pair there is less possibility of leakage of charge through the exhaust port. The exhaust is closed before the transfer port, and owing to their differential movement at the top of the stroke (during ignition) the gases rush through the port connecting the two cylinders and much beneficial turbulence should result.

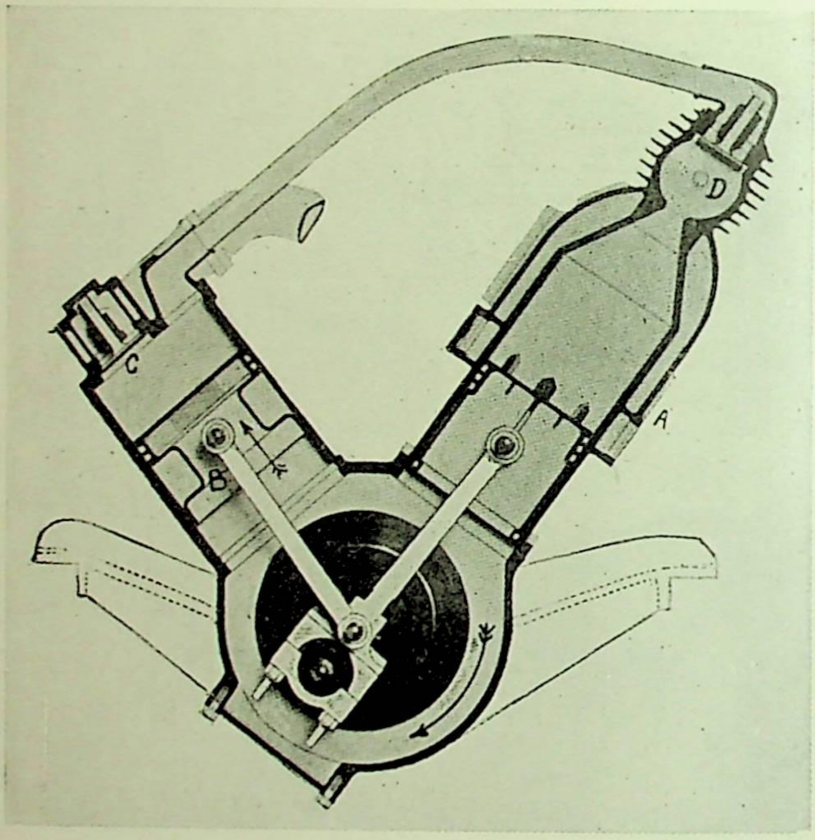

Fig. 165.—Two-stroke Engine using Separate Compression Cylinder.

This engine, which is designed to give its maximum power over a range of speeds, has of course a flat power curve and it is claimed that this engine gives 10 h.p. at 400 and at 1,200 r.p.m., so that it is suitable for a top-gear car. The engine contains only seven moving parts, of which the connecting rods are Vee-shaped, one double rod serving for the two pistons of a cylinder pair.

SEPARATE COMPRESSION CYLINDER

Another type of engine employs a separate (and usually inclined) cylinder for compressing the fresh charge, which is then forced through a non-return valve into the head of the main cylinder, when the main piston therein is at the bottom of its firing stroke. The piston of the subsidiary cylinder on its downstroke draws into the cylinder a mixture of air and petrol from the carburettor, which it compresses in the cylinder until at a certain predetermined pressure a spring-loaded valve opens and admits this charge into the firing cylinder. It will be obvious that by making the capacity of the subsidiary cylinder sufficiently large, the mixture admitted into the main cylinder will be sufficient to completely fill it under a pressure higher than would be the case with crankcase compression only.

The objection to charge loss through the exhaust port has been overcome in the case of a certain class of petrol engine by utilising the crankcase, or alternatively a separate piston and cylinder, for compressing pure air alone, which latter is then admitted in the usual manner through an inlet port, whilst the exhaust port is uncovered. The piston on its upward stroke compresses the pure air, and when it is almost fully compressed the petrol, or other fuel, is sprayed into the cylinder head under pressure, and in suitable amount to give the correct mixture-strength. Any loss which occurs through the exhaust port is therefore only one of air, and not fuel.

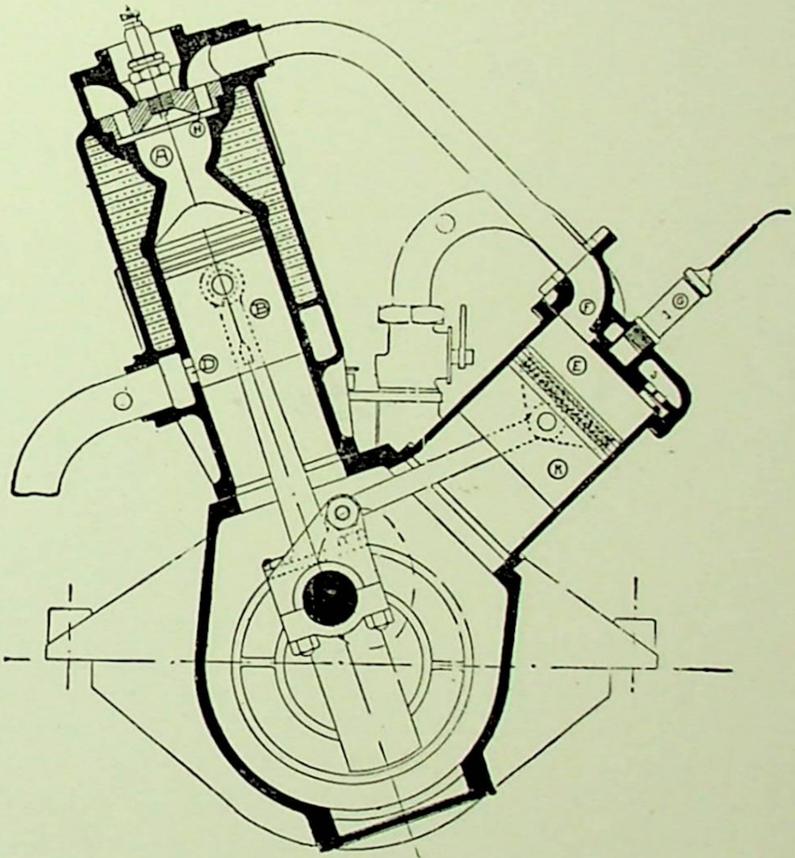

Fig. 166.—Section of Two-stroke Vee-engine.

This type is usually known as the *solid injection* engine, and is very widely employed in heavy-oil and Diesel-type engines. There are favourable indications of its future use in both aircraft and automobile engines.

An alternative method of compressing the charge, and one which has frequently been employed, is that of the differential piston, in which the piston is of larger diameter below, and this portion works in a corresponding projection of the cylinder. The compression occurs within the annular space between the trunk of the smaller part of the piston and the larger cylinder wall. In this way any degree of compression can be obtained. This form of construction results in a compact engine, but of rather greater overall height than in the ordinary type; the stepped piston is also rather heavier than usual.

OIL ENGINES

For motor-transport purposes, more particularly in the case of the heavier types of vehicle, the oil engine possesses certain useful features which recommend its use.

In the first place this engine generally runs upon paraffin fuels, which work out cheaper than petrol and benzol; it also runs at a somewhat slower speed, and can be built so as to work for long periods with little attention. In the slow-speed type of oil engine the speeds vary from about 75 to 250 r.p.m., and in the high-speed class from about 250 to 1,200 r.p.m. The range of speeds in the case of the ordinary petrol engine is about 300 to 3,000 r.p.m., but the high-speed engines run up to about 4,000 r.p.m. These engines usually work on the Otto, or four-cycle, and the Clerk, or two-cycle principles, and differ from the petrol engine in the method of introducing the mixture into the cylinder and in the means of ignition. The compression pressures are also much higher than in the case of the petrol engine.

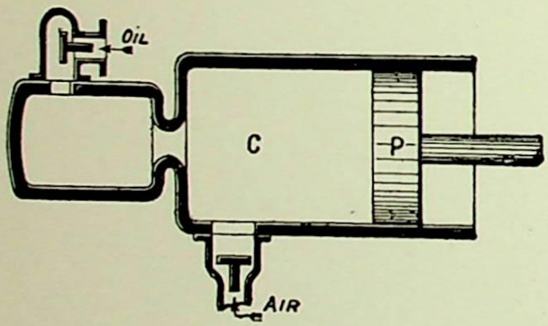

Fig. 167.—Spontaneous Ignition Vaporiser.

There are many different types of oil engine, some working almost identically with the petrol engine in the method of introducing the oil-air mixture. Others compress the air only and inject the oil during the compression stroke and employ hot-tube ignition. In a third class the combustion chamber has a small outlet leading to a small, nearly spherical chamber, which is initially heated with a blow lamp, and once it fires, this chamber is maintained hot enough to ignite each succeeding charge at the right moment by the heat of explosion. In a fourth the fuel is pumped into a small passage leading from the cylinder to a vaporiser. The gas pressure produced by a small fraction of the charge in the vaporiser injects the main charge into the combustion chamber.

The fuel may be injected into the cylinder either by the gas-pressure method just described, or by compressed air at a greater pressure than the maximum compression pressure, or by mechanical means such as by pumping.

An example of the automatic-ignition oil engine is shown diagrammatically in Fig. 167. Here the oil and air are sprayed into a hot chamber on the cylinder end and are then compressed to a certain predetermined pressure ; the temperature of the charge increases with the pressure until at the given value the temperature is sufficient to ignite the charge. The temperature of ignition will depend upon the kind of oil used, and so the compression must be adjusted accordingly. The *hot-bulb* type of engine described has proved very satisfactory in practice. It is often combined with an electric-ignition device for starting purposes, petrol being used until the engine warms up.

Figs. 169 and 170 illustrate the design of the oil injector of an oil engine, working upon the compressed-air principle. Here, a needle valve is arranged in the cover of the cylinder head and is depressed by means of an elbow lever and cam. During the opening period of the fuel needle valve compressed air sprays the fuel into the cylinder, and it fires, or combusts, as it enters, owing to the high temperature due to the compression. Pressures of 500 lb. per sq. inch and above are used for compression, and the temperatures attained are sufficient to ignite the injected fuel. In this case the air only is compressed beforehand in the cylinder.

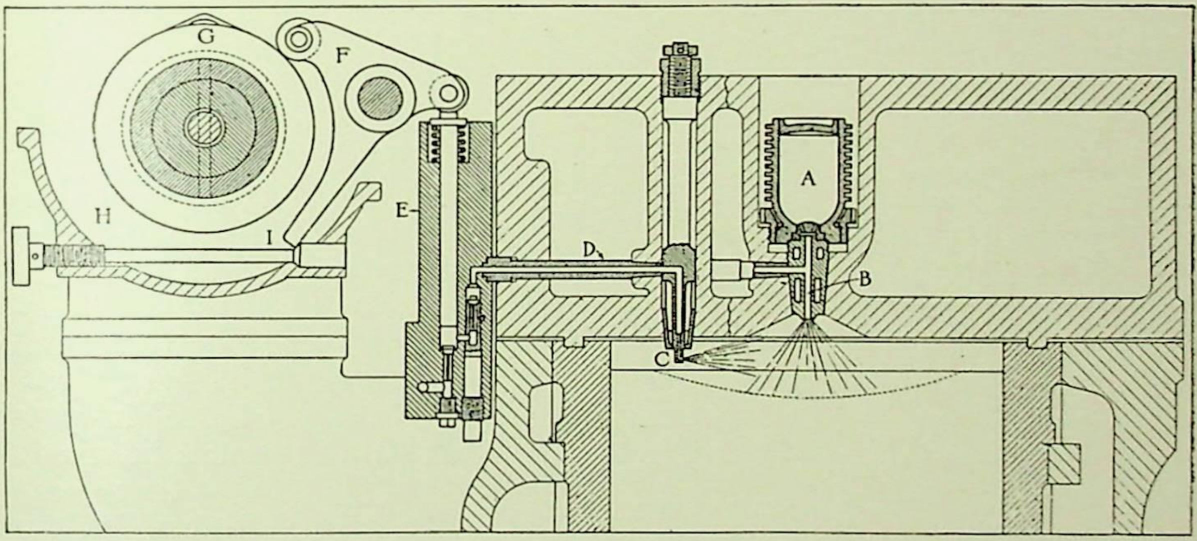

Fig. 168.

This is the principle of the Diesel engine, and this type will work upon crude shale and petroleum oils of a tarry consistency. Already these engines have been employed upon locomotives and heavy commercial vehicles with satisfactory results.

An engine of the single-cylinder, two-piston type has been built in Germany for automobile purposes, having a cylinder of 130 mm. bore and 160 mm. stroke, giving a combined stroke of 320 mm. for the two pistons. The output of this engine is 35 b.h.p. at 850 r.p.m. The weight of Diesel engines has hitherto been rather excessive, for the stationary types weigh from 200 to 500 lb. per b.h.p., but light sets of small horse-power have been built weighing only 40 to 50 lb. per b.h.p. In the above example the weight has been brought down to 22 lb. per b.h.p., and there is no reason why the weight should not be reduced still further. The average motor-car engine weighs from 8 to 12 lb. per b.h.p. in the water-cooled model, with a plentiful use of aluminium in its construction, and from 4 to 8 lb. in the case of modern air-cooled cycle-car-type engines with aluminium and steel cylinders.

Fig. 168[1] illustrates a section through the cylinder head of the Diesel engine previously referred to, and will serve as an example of the action of this type of engine. The fuel-injection system used is known as the *Steinbeker,* and it avoids the use of air compressors, furnishing air up to 1,000 lb. per sq. inch for injection purposes. There is a small vessel A, termed the "ignition chamber," attached above the cylinder head, and it communicates with the combustion chamber through a central passage called the *firing passage.* The cylinder is provided with the usual inlet and exhaust valves. There are two fuel jets, B and C respectively, of which the former delivers fuel into the firing passage and the latter direct into the cylinder. The fuel itself is forced under pressure by means of a pump E, of the positive type, provided with the two non-return valves shown, and operated by means of the three-armed lever F, upon one end of which is a roller bearing upon the cam G. The quantity of fuel injected is determined by the stroke of the pump, and this in turn can be varied by altering the position of the stop I, on one of the three arms of the lever F, by means of the screw H.

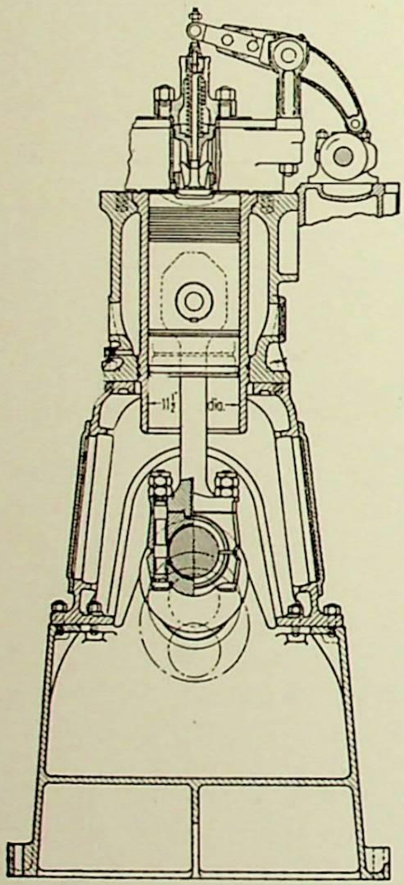

Fig. 169.

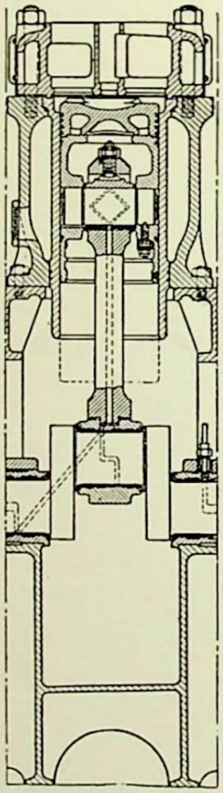

Fig. 170.

The load, or output, of a Diesel engine is always governed in this manner, that is, by regulating the amount and period of the fuel injection.

Reverting to the example illustrated in Fig. 168, the action of the engine is as follows: During the inward stroke of the piston, pure air is drawn into the cylinder, and this is compressed on the outward stroke, during which process some enters the chamber A, the final pressure being of the order 400 to 500 lb. per sq. inch. At the end of this stroke fuel is injected into the firing passage and is carried into the firing chamber, which is sufficiently hot to ignite the mixture formed. The pressure at once rises to about 1,100 to 1,400 lb. per sq. inch in the firing chamber, and as a result the flame is projected into the combustion chamber, carrying with it fuel from the firing chamber and spraying it into the combustion chamber. When the piston has travelled a short distance down its stroke the fuel supply is cut off, and the exceedingly hot ignited charge expands until the bottom of the piston's stroke is reached. The gases are ejected through the exhaust valve on the upward stroke of the piston. The column of flame which is projected from the ignition chamber A of course ignites the fuel entering through the jet C. In this manner, the pressure due to the partial explosion of the small charge in A takes the place of the ordinary high-pressure air used in the ordinary Diesel practice for injection of the fuel.

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[1] By courtesy of "Automotive Industries."

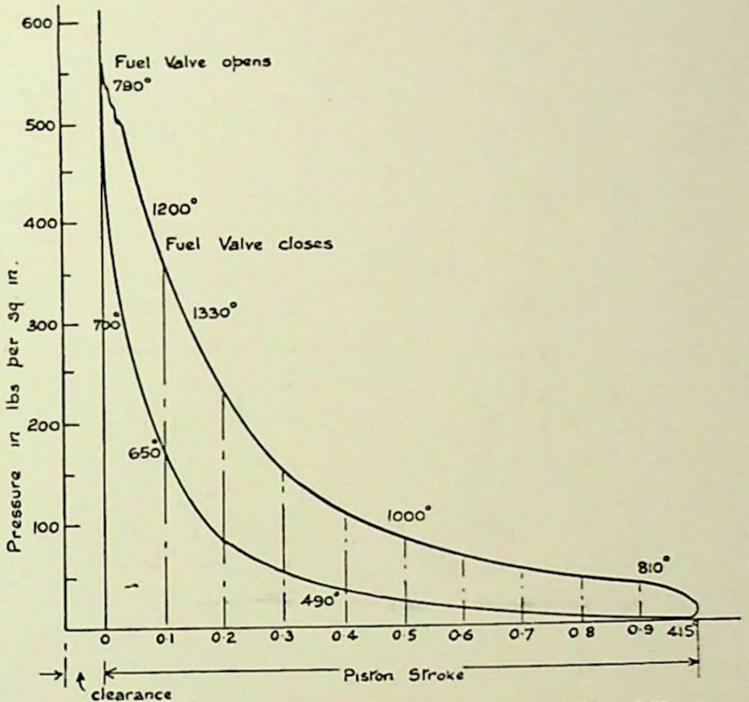

Fig. 171.—Pressure and Temperature Diagram of Diesel Engine.

THE TARTRAIS OIL ENGINE

Much attention has been given to the subject of oil engines for automobile work by French engineers, and as a result a very successful type of two-stroke heavy-oil engine has been made by the Peugeot Company under the Tartrais patents, and has carried out some satisfactory road tests in comparison with petrol-type engines made by the same firm.

Two cars, identical in every respect, except for their engines, were driven for a whole day, at various road speeds, each doing over 200 miles. The Tartrais-engined car performed equally well with the petrol-engined car, although rather slower in maximum acceleration and hill-climbing speed, but it showed a marked superiority in fuel consumption and cost. The oil used worked out at 19·88 miles per gallon, whereas in the case of the petrol engine the petrol mileage per gallon was 18·87. The fuel oil used, namely crude gas oil, cost only 0·426 pence per mile, whereas the petrol used worked out at 1·14 pence per mile.

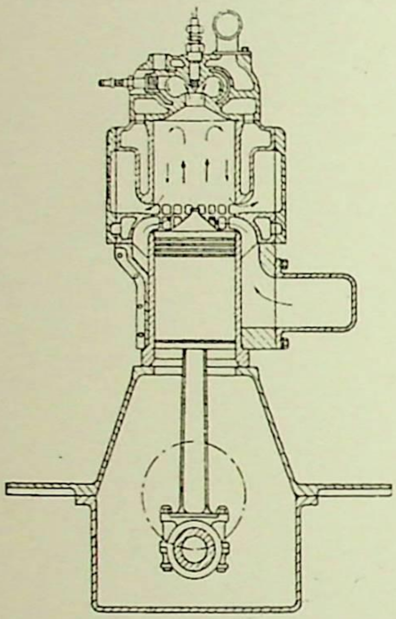

Fig. 172.—The Tartrais Oil Engine.

Figs. 172 and 173[1] illustrate the Tartrais two-cylinder high-speed oil engine in section; the bore and stroke are 120 mm. and 150 mm. respectively. This engine works on the two-stroke principle, the pistons uncovering exhaust ports in the usual manner. At the top of the combustion head is an annular ignition chamber which communicates with the combustion chamber, but the communication passage is partially closed by a projection on the head of the piston when the fuel injection takes place. This arrangement produces much turbulence and renders the combustion process very efficient, as is shown by the low fuel-consumption values, namely 0·4 lb. per b.h.p. hour. The ignition chamber is uncooled and therefore becomes very hot soon after the engine starts up (a special starting plug being employed, in conjunction with petrol, for starting purposes); this chamber becomes sufficiently hot to ignite the fuel-air mixture. Pure air is compressed to about 5 lb. per sq. inch in a separate compressor (shown on the left of Fig. 173), and is admitted to the cylinders through ports situated below the exhaust ports, so that a certain scavenging effect is obtained. The ascending piston then compresses this air, and when the piston reaches the top end of its stroke this air is forced at a very high velocity through the very narrow annular passage, past the projection on the piston top and into the ignition chamber. The compression pressure is about 280 lb. per sq. inch, and the maximum explosion pressure from 400 to 450 lb. per sq. inch.

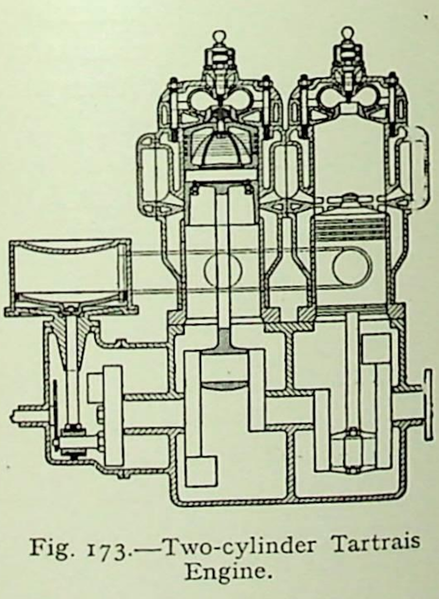

Fig. 173.—Two-cylinder Tartrais Engine.

The weight of this engine, which is of 35 h.p., and which works at 1,400 r.p.m., is only 11 lb. per b.h.p., so that it compares very well with that of modern petrol engines.

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[1] *Inst. Aut. Engr. Proc.* 1923.

CHAPTER IV

THE LUBRICATION SYSTEM

ONE of the secrets of the success of the modern automobile engine is to be found in connection with the high standard of efficiency of the lubricating systems in use. The loads and speeds of rubbing of the sliding surfaces are fairly high in petrol-engine practice, so that the question of adequate lubrication becomes one of much importance, from the points of view of reliability over long periods, and of minimum wear.

The function of the lubricating oil is to provide a liquid film between the two bearing surfaces (e.g. the crank journal and the big-end bearing), so as to reduce the friction from the "solid" to the "fluid" type. The friction of a well-lubricated bearing is then merely the fluid resistance or drag of the oil film, since the two metal surfaces do not touch. Immediately the load between the two parts forming the bearing becomes too great or the oil too thin (through heat or insufficient viscosity), the oil film is destroyed and the two metal surfaces seize together.

When a journal rotates in a well-lubricated bearing the oil film in the best practice will be of the order 0·0007 to 0·0002 inch and the frictional resistance is then found to be proportional to (*a*) the speed, (*b*) the area, (*c*) the viscosity of the lubricant. It is independent of the pressure between the journal and bearing, provided that this is not so great that the oil film breaks down, and it is also independent of the material of the surfaces. The amount of oil which is supplied to a bearing is also important, for the supply must make up for the loss of oil due to its escape from the ends of the bearing ; on the other hand, it is not advisable to flood the bearings of a petrol engine, or the oil will find its way past the piston and will become "burnt," so that the engine will smoke considerably. In time, excessive deposit will be formed on the piston head and combustion chamber, and the exhaust-valve stem will become coated with a tarry deposit, so that after a comparatively short period of running the engine will lose power and work erratically.

LUBRICATING OILS

There are two kinds of oil used for engine lubrication, of mineral and vegetable origin respectively. The former class, which includes the hydrocarbon and compound oils, is almost universally employed, whilst the latter class is now represented by castor oil, which is limited to high-speed, racing, and certain types of aircraft engine.

The principal properties upon which the lubricating value of an oil depends are: (1) its viscosity at the working temperatures of the engine; (2) its flash point; and (3) its heat loss—that is, the percentage of oil lost on heating for a period below the flash point.

The viscosity of an oil depends upon its constitution, and falls off fairly rapidly as the temperature is raised. A good engine oil should not be too viscous when cold, otherwise it will be difficult to crank the engine over; neither should it be too "thin" at the cylinder wall temperature, or it will be squeezed out from between the bearing surfaces and scoring of the walls will result. The ideal oil would have the same viscosity under all conditions, but unfortunately this is not possible.

The smaller the change in viscosity of the oil, the better, since the engine pump should keep up a constant supply, and it is the viscosity which opposes the oil-flow in the pipes; for this reason a thick oil is at a disadvantage.

The flash point of an oil is the temperature at which it gives off inflammable vapour (i.e. which burns in air). Ordinary lubricating oils usually have a flash point of from 400° to 500° Fahr., and are therefore quite safe, even under tropical conditions.

There is another property of lubricating oils which is of very great importance, namely their degree of "oiliness" or "greasiness." It is known that two oils may have the same flash point and viscosity at the engine temperatures, but may differ considerably in their lubricating properties; there appears to be no convenient physical or chemical explanation of this property.

The effect of temperature upon viscosity, which has been mentioned before, is very important, but it is known that the differences in viscosity are comparatively slight at the higher temperatures. Vegetable oils do not fall off in viscosity to the same extent as mineral oils; in the case of animal oils, such as sperm or lard oil, the rate is also not so rapid.

Castor oil possesses this property, and it still possesses a fairly good viscosity and a better "oiliness" than mineral oils at the high cylinder temperatures attained in motor-cycle and racing-car engines, and it is therefore much used in these cases. Another advantage of castor oil is that it is not readily soluble in petrol, whereas mineral oil is. In the case of partly worn engines some of the petrol finds its way into the crankcase and causes the lubricating oil to become thinner, although there is another reason for this, namely, the exposure to the high cylinder temperatures. Vegetable oils are known to be bad carbon-formers, in petrol engine cylinders.

In the case of air-cooled engines, the cylinder temperatures are much higher than in water-cooled ones, so that a lubricating oil of a higher viscosity (a thicker oil) is required. Frequently a motor-cycle engine is partially " cooled " by the lubricating oil, as is also evident from the fact that an extra pumpful of oil is required on hills, etc.

The grade of oil employed in the summer, or in hot countries, is such that the viscosity is much higher than in the case of oils used under winter conditions. Most reputable makes of oil are sold in the " summer " and " winter " grades.

The particular kind of oil necessary for any given engine will depend a good deal upon the design, type, and conditions of running. Generally speaking, an engine which normally works at a high speed, or one with inefficient cooling means, or with pressure-fed lubrication, and air-cooled types, and require an oil of the high-viscosity or " thick " grade ; whereas a low-speed engine, an over-cooled one, or a splash-lubricated one would require a low-viscosity or " thin " grade. In both cases only the best recognised brands of lubricating oil are assumed. In lubrication adjustments, the golden rule to observe is " to oil rather on the generous side with the best lubricant " ; at the worst, over-oiling only means more frequent decarbonising of the engine, whereas under-oiling may easily result in a ruined engine.

LUBRICATING SYSTEMS

The parts of a petrol engine which require lubricating during the running period are the piston and cylinder rubbing surfaces, gudgeon-pin bearing, connecting-rod big-end bearing, main bearings, valve-timing gearing, camshaft cams and bearings, the valve tappets and guides, and the auxiliary drive bearings. The majority of these parts are enclosed, but outside parts such as valve-rocker bearings, pump and fan drives, etc., require additional attention, at intervals.

Engine lubrication systems may be divided broadly into three classes as follows : (1) the splash system ; (2) the splash-pressure system ; and (3) the pressure or forced-feed system. These will be dealt with in the order enumerated, before passing on to a consideration of the different types of pump used, and other details of lubricating system.

The splash system is the one which is more widely employed than any other, and, owing to the attention which has been given to the details of its design, it is not only fairly simple and therefore cheap to embody, but is also thoroughly reliable. It has certain disadvantages, but these are not serious ; reference will be made to these later.

The splash method consists in placing troughs immediately under the lowest part of the travel of the connecting rods, and in keeping these troughs filled with oil. Scoops, or dippers, in the ends of the connecting rods (similar to the ones shown in Fig. 97) dip periodically into these troughs and supply oil not only to the big ends, but also throw the oil in different directions, including the cylinder walls. The excess oil usually runs down the sides of the crankcase and is caught by troughs over the main bearings, provided with oil holes. The oil, which is rapidly scooped up and flung about, becomes virtually an oil-spray or cloud and settles upon every exposed surface.

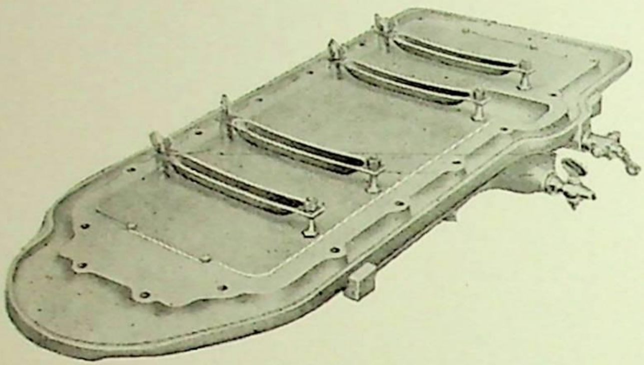


Fig. 174.—The A.E.C. Engine Lubrication Troughs.

The oil troughs are usually rectangular in form, as shown in Fig. 174, which represents the A.E.C. commercial engine crankcase oil sump. In this example it will be observed that winged nuts (accessible from doors in the crankcase) are fitted for adjusting the levels, or heights, so as to regulate the lubrication.

These troughs are kept full of oil by means of an oil-pump which draws oil from a tank, or sump, below the level of the troughs—usually the base of the crankchamber, as shown in Fig. 180—and pumps it directly through pipes feeding the troughs, or to the main bearings first, whence it escapes to a base containing the troughs, any excess of oil being returned through an open-ended overflow pipe into the sump.

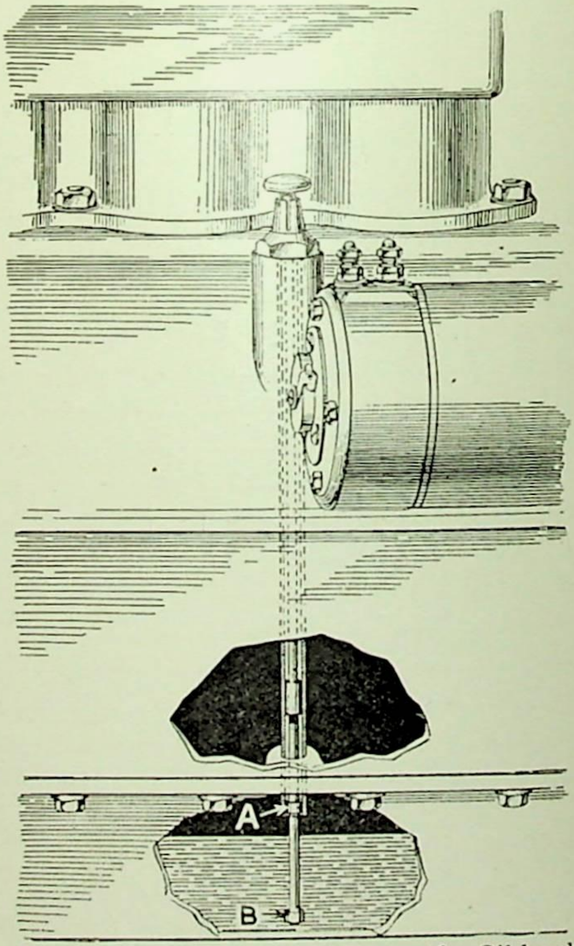
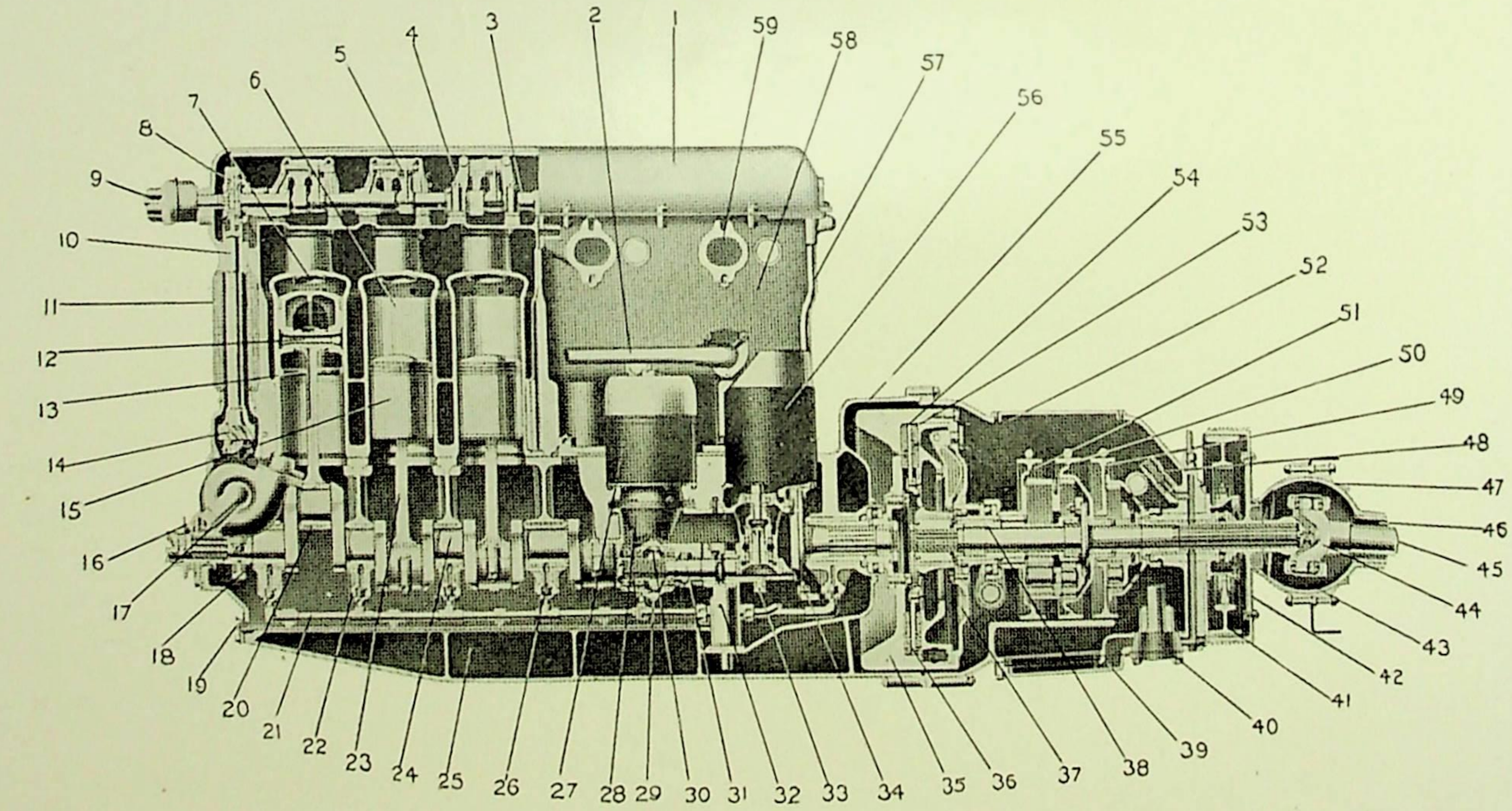


Fig. 175.—The Dipper-rod System for Oil-level Indication.

The sump, which usually carries from one-half to one gallon of oil in the case of a 12 h.p. engine, is provided with an oil-supply pipe and gauze strainer. A float type of oil-level indicator is usually fitted upon the side of the sump in the case of the better types of car engine, although the dipper-rod method employed on the 12 h.p. Austin engine (Fig. 175) is equally efficient.

A common method for determining the level of the oil in the sump is to fit an ordinary cock or tap of ample bore at the correct oil level; by opening the tap at any time it can be seen whether the oil level is above or below this tap. An improvement upon this method is to fit two taps in the side of the oil sump, one at the maximum advisable level and the other at the lowest safe level to which the oil should fall;



The Lanchester Six-cylinder Engine in Section.

1. Valve Cover.
2. Water Circulation Pipe.
3. Camshaft.
4. Camshaft Bracket.
5. Valve Tappet Lever.
6. Cylinder.
7. Inlet Valve.
8. Camshaft Worm Wheel.
9. Distributor.
10. Vertical Shaft.
11. Vertical Shaft Casing.
12. Gudgeon Pin.
13. Connecting Rod Oil Conduit.
14. Magneto Drive Shaft Worm Wheel.
15. Piston.
16. Fan Pulley.
17. Centrifugal Water Pump.
18. Vertical Shaft Worm Wheel.
19. Oil Gauze Cover.
20. Connecting Rod Big End Bearing.
21. Oil Pipe.
22. Crankshaft Main Bearing Cap.
23. Connecting Rod.
24. Crankshaft.
25. Oil Base.
26. Crankshaft Bearing Cap Oil Way.
27. Starter.
28. Starter Drive Housing.
29. Starter Worm Wheel.
30. Starter Free Wheel Clutch.
31. Layshaft.
32. Engine Oil Pump.
33. Generator Worm Wheel.
34. Layshaft Driving Wheel.
35. Flywheel.
36. Clutch Disc.
37. Clutch Cover Plate.
38. Gear Box Centre Shaft.
39. Gear Box Oil Sump.
40. Countershaft Oil Pump.
41. Countershaft Brake Shoe.
42. Countershaft Brake Drum.
43. Torque Tube Ball.
44. Universal "Hooks" Joint.
45. Cardan Shaft.
46. Torque Tube.
47. Torque Tube Spherical Joint.
48. Oil Pressure Relief Valve.
49. Compound or Second Gear Drum.
50. Low Gear Drum.
51. Reverse Gear Drum.
52. Gear Box.
53. Fabric Clutch Disc.
54. Fabric Clutch Disc.
55. Flywheel Casing.
56. Generator
57. Oil Pipe.
58. Cylinder Water Jacket.
59. Exhaust Port.

In the case of some engines the oil is contained in a plain tray or trough, provided with an overflow pipe, this being a simple, inexpensive construction. The disadvantage of this method is that on an incline the end cylinder (nearest the lowest part of the incline) receives considerably more oil than the others, as shown in Fig. 177, A. With the separate trough system this does not occur, although there may be a partial draining of the oil troughs. Fig. 177, B shows the latter method.

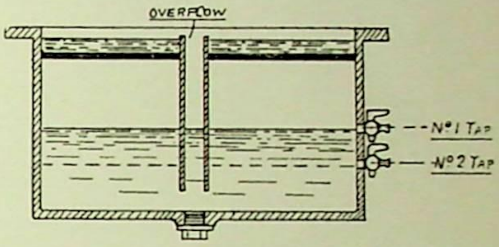


Fig. 176.

In the case of the Daimler sleeve-valve engine the regulated splash system of lubrication is adopted. Separate troughs are provided, into which the connecting rods dip in the usual manner and a multiple plunger oil pump, together with a mechanically operated valve controlling the suction and delivery, draws oil from the sump through a strainer and delivers it through pipes into the troughs, which are kept overflowing with oil. As the crankshaft revolves, the connecting-rod scoops throw the oil on to the pistons and sleeves. The troughs are mounted on hinges, as shown in Fig. 178 (at the back), and are interconnected with the throttle in such a way that the extent to which the connecting rod dips into the oil increases as the throttle is opened. The illustration shows the six cast-aluminium troughs with their oil-feeding pipes, and also shows the operating lever attached to the long connecting rod passing through the front ends of the troughs. The operating lever in the upper centre of the crankcase shown in the photograph is connected with the carburettor throttle ; the connection can also be seen in Fig. 139. The lubrication is thus varied automatically to suit the engine power.

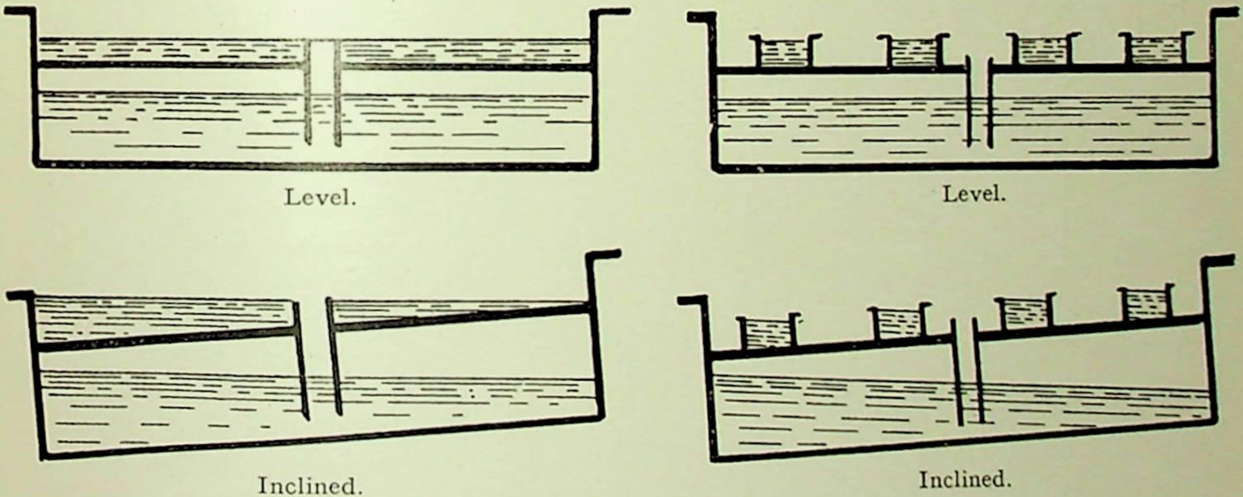


Fig. 177.

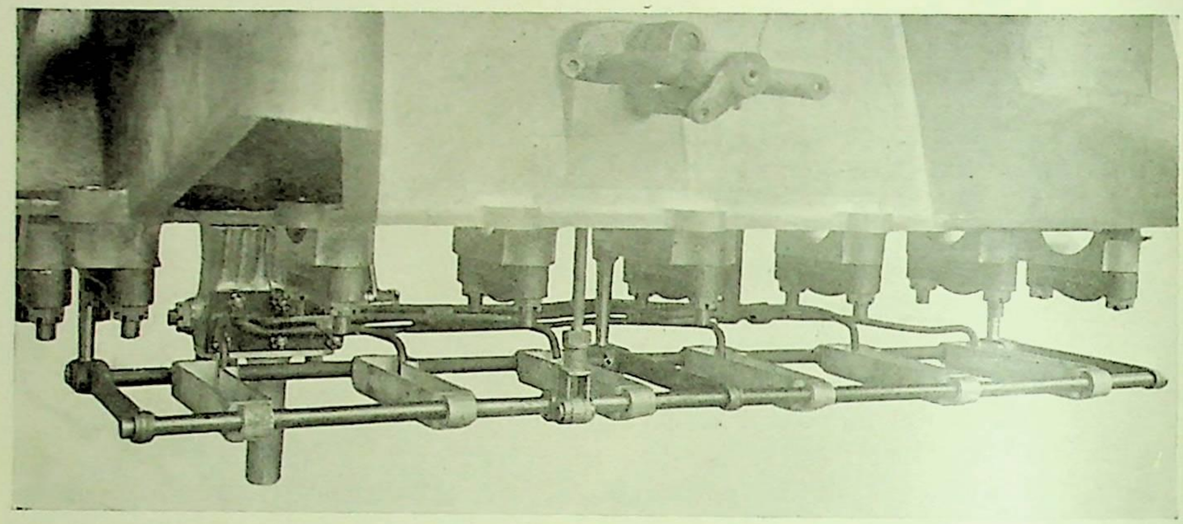


Fig. 178.—Illustrating the Daimler Hinged Oil Troughs, connected to the Throttle Lever.

Fig. 179 shows the complete Daimler system of lubrication. In addition to supplying the troughs, the pump also feeds oil to channels communicating with the main bearings of the crankshaft and to the chambers containing the chain drive for the sleeves. This oil supply is maintained by the central chamber of the multiple pump, and the oil is passed through another fine gauze filter. The cap that covers the orifice through which the oil is poured into the crank chamber is connected to a cock situated at a certain level from the bottom of the crank chamber. When the cap is moved to open the oil-filled hole, this oil-level cock is also

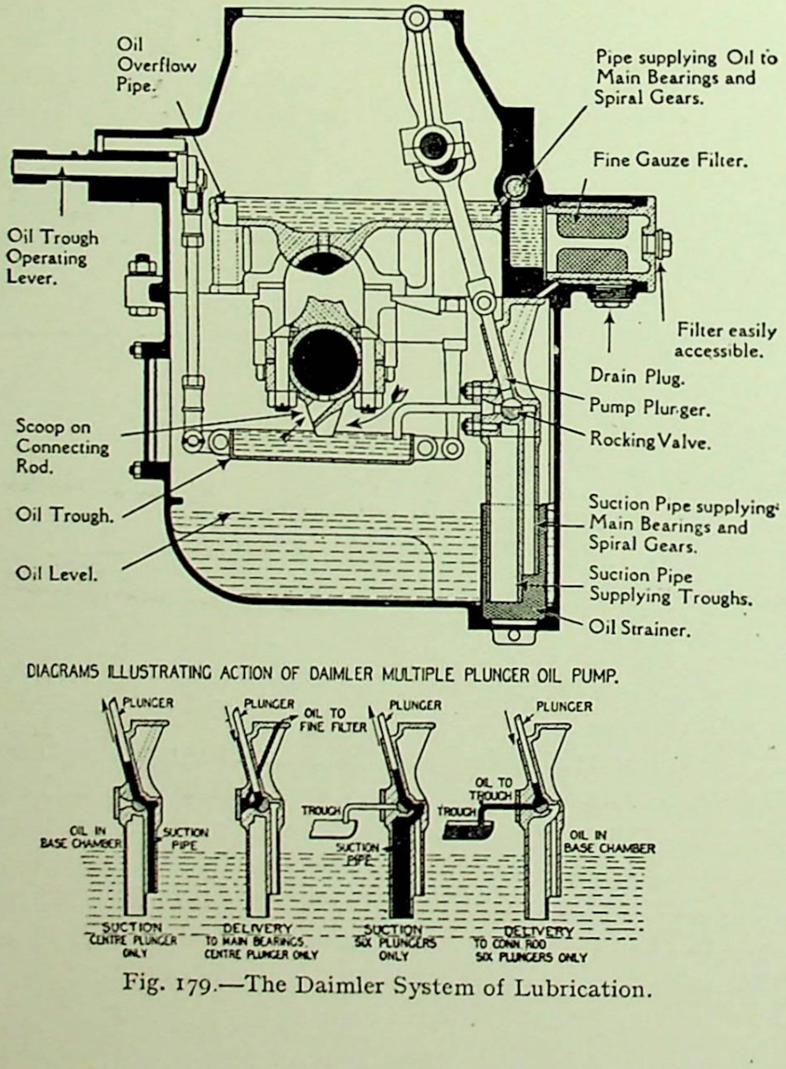

Oil Overflow Pipe.
Pipe supplying Oil to Main Bearings and Spiral Gears.
Fine Gauze Filter.
Oil Trough Operating Lever.
Filter easily accessible.
Drain Plug.
Pump Plunger.
Rocking Valve.
Scoop on Connecting Rod.
Oil Trough.
Suction Pipe supplying Main Bearings and Spiral Gears.
Oil Level.
Suction Pipe Supplying Troughs.
Oil Strainer.
DIAGRAMS ILLUSTRATING ACTION OF DAIMLER MULTIPLE PLUNGER OIL PUMP.
PLUNGER
PLUNGER
OIL TO FINE FILTER
PLUNGER
PLUNGER
OIL TO TROUCH
TROUCH
TROUCH
OIL IN BASE CHAMBER
SUCTION PIPE
SUCTION PIPE
OIL IN BASE CHAMBER
SUCTION CENTRE PLUNGER ONLY
DELIVERY TO MAIN BEARINGS CENTRE PLUNGER ONLY
SUCTION SIX PLUNGERS ONLY
DELIVERY TO CONN. ROD SIX PLUNGERS ONLY

Fig. 179.—The Daimler System of Lubrication.

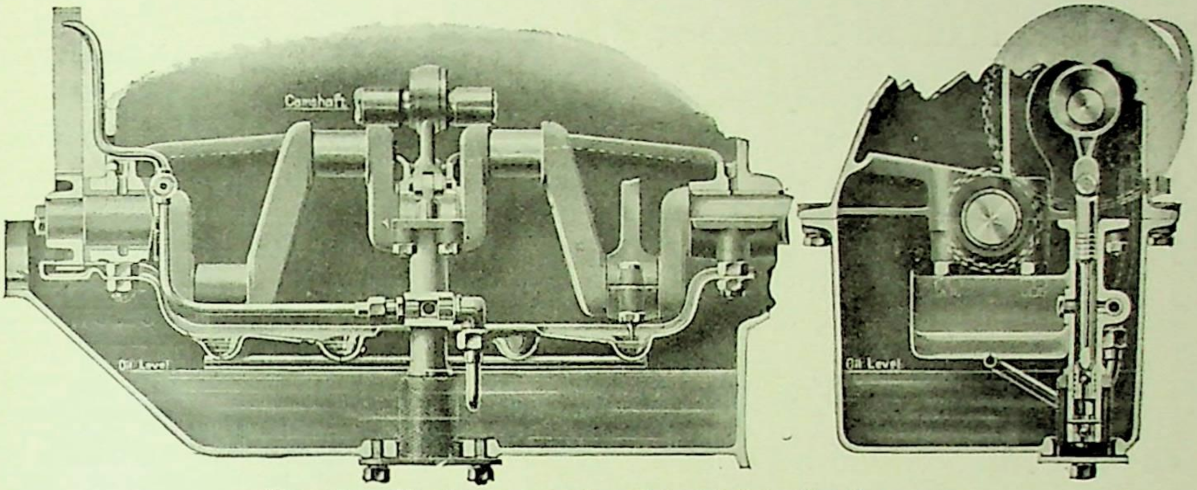


Fig. 180.—The Dorman 12 h.p. Splash-lubrication System.

opened, and when oil flows therefrom it is a sign that sufficient oil is already in the crank chamber. The oil should be replenished in these cases when the car is on a level road, and after every 1,500 to 2,000 miles it is advisable to wash the sump out with paraffin, and also to clean the oil strainers.

The lubrication system of the 10 h.p. Dorman engine, described on p. 77, is shown in Fig. 180. It is on the splash system, and the illustrations show how this is carried out. The view on the right-hand side shows the eccentric-driven pump, which is of the plunger type, and is

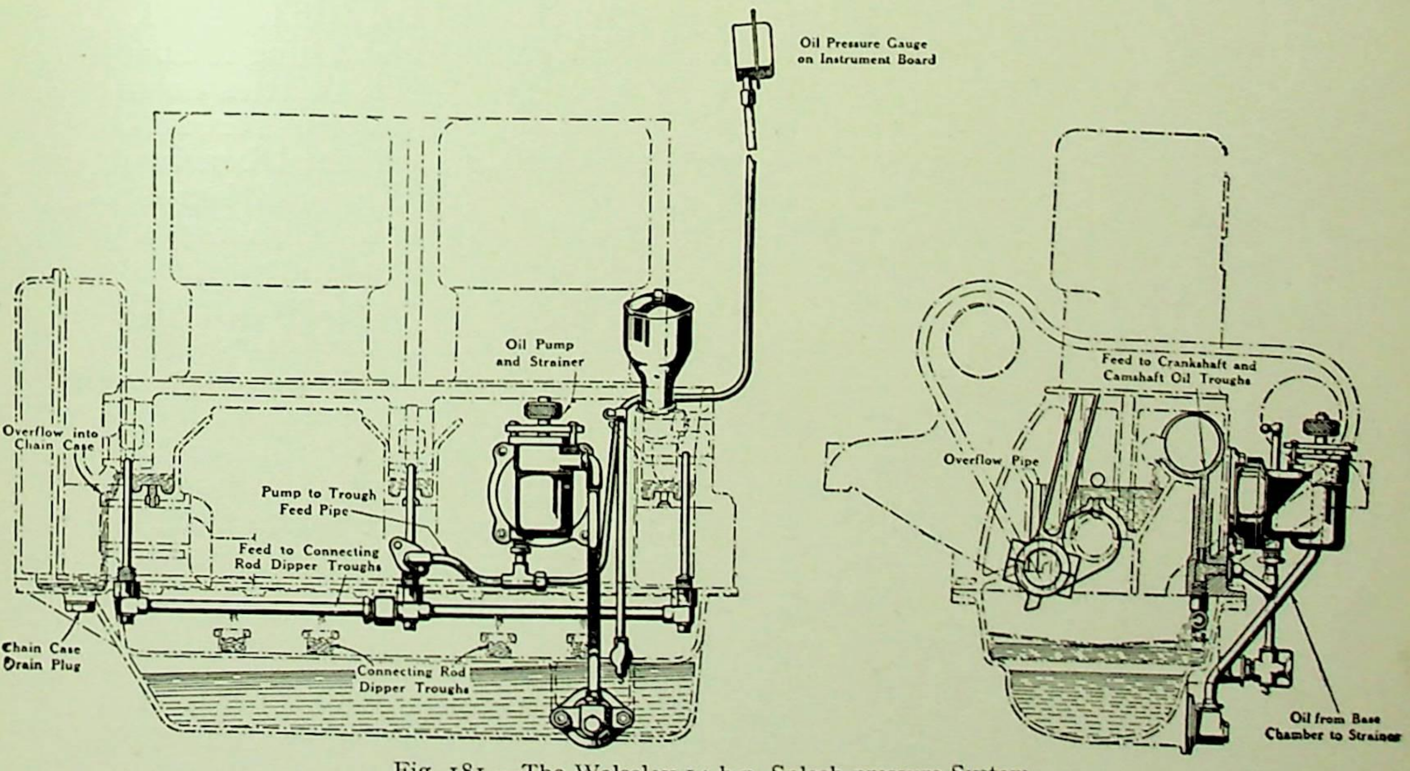

Oil Pressure Gauge on Instrument Board
Oil Pump and Strainer
Overflow into Chain Case
Pump to Trough Feed Pipe
Feed to Connecting Rod Dipper Troughs
Chain Case Drain Plug
Connecting Rod Dipper Troughs
Feed to Crankshaft and Camshaft Oil Troughs
Overflow Pipe
Oil from Base Chamber to Strainer

Fig. 181.—The Wolseley 14 h.p. Splash-pressure System.

submerged in the oil of the sump at its lower end. The plunger on its upward stroke draws oil from the sump through a wire gauze strainer secured to a cage surrounding the intake, as shown in the centre of the left-hand view in Fig. 180 ; on the downward stroke it discharges this oil upwardly past another ball valve and through its casing to two distributing pipes, one leading to troughs under the big ends and the other to a point just above the timing gears. A branch is arranged for a lead to an oil-pressure indicator on the dashboard. The pump plunger has two long bearings widely spaced, and the gauze filter is removable from below for cleaning purposes. Pressed-steel troughs in an integral unit, or pressing, are employed in this system, and the troughs have their rear walls higher than those in front, in order to maintain the oil level when the car is climbing hills. There is also a fifth trough provided for the crankshaft timing pinion to run in. The oil is fed into the troughs from below, baffles being provided to prevent the oil from surging or squirting upwards out of the trough.

MOTOR-CYCLE LUBRICATION

At this stage it may be of interest to describe the usual methods whereby motor-cycle engines are lubricated.

The simplest (and crudest) method is to lead a pipe from the oil tank to a point near the base of the crank-case, the oil being fed by a drip feed on the tank, with an auxiliary hand pump for forcing the oil through the drip feed. The flywheels dip into the oil and fling it about in the form of a spray which settles on the working parts ; oil holes and leads are fitted to direct the oil to the principal bearings ; holes are drilled through the wall between the timing and the crank chambers, for oil to find its way to the timing gear.

In the case of a Vee-twin engine, this method usually results in the rear cylinder being over-lubricated, since it is the first to receive the oil thrown upwards and outwards off the flywheel. To obviate this, baffles are usually fitted over the lower end of the rear cylinder, with a slot for the rear connecting rod to work in. The modern tendency, however, is towards mechanical lubrication, with a pump to draw oil from a sump and force it to the main bearings, big ends, and timing gears, whence it drains back into the sump.

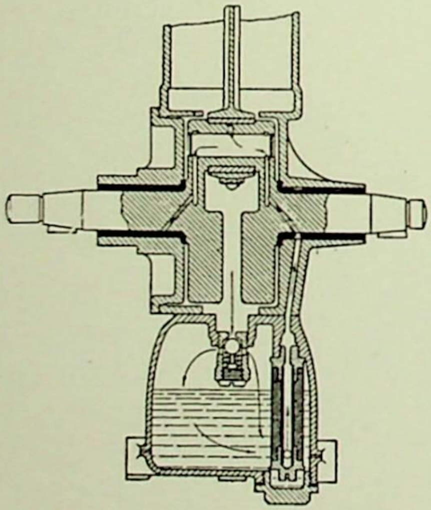


Fig. 182.

Fig. 182 illustrates what is known as the *dry-sump* method of lubrication for motor-cycle engines. Here the oil drains through a ball valve into the sump, and the pressure in the crankcase due to the descending piston acts on the surface of the oil in the sump and forces the oil up the channel on the right, through the main bearings and holes drilled in the crankshaft to the hollow crankpin and big-end bearing. Alternatively, a plunger pump can be used as previously mentioned.

In the case of many two-cycle engines the *petroil* system is employed and gives very satisfactory results. It consists in mixing about one part of lubricating oil to every twelve to sixteen parts of petrol, and feeding this oily petrol through the carburettor in the usual manner. The oil is thus fed into the crankcase, and thence to the cylinder, and the amount of oil is thus proportional to the load.

THE SPLASH-PRESSURE SYSTEM

In this system oil is fed direct to the main bearings from the pump under pressure, as distinct from the method of *free* feed to the bearings from open-ended pipes above the bearings. The oil in the present system travels through the drilled crankshaft to the big-end bearings, under the influence of the pressure, and by centrifugal action. The piston and gudgeon pins are lubricated by the oil splashed about and thrown off the big ends.

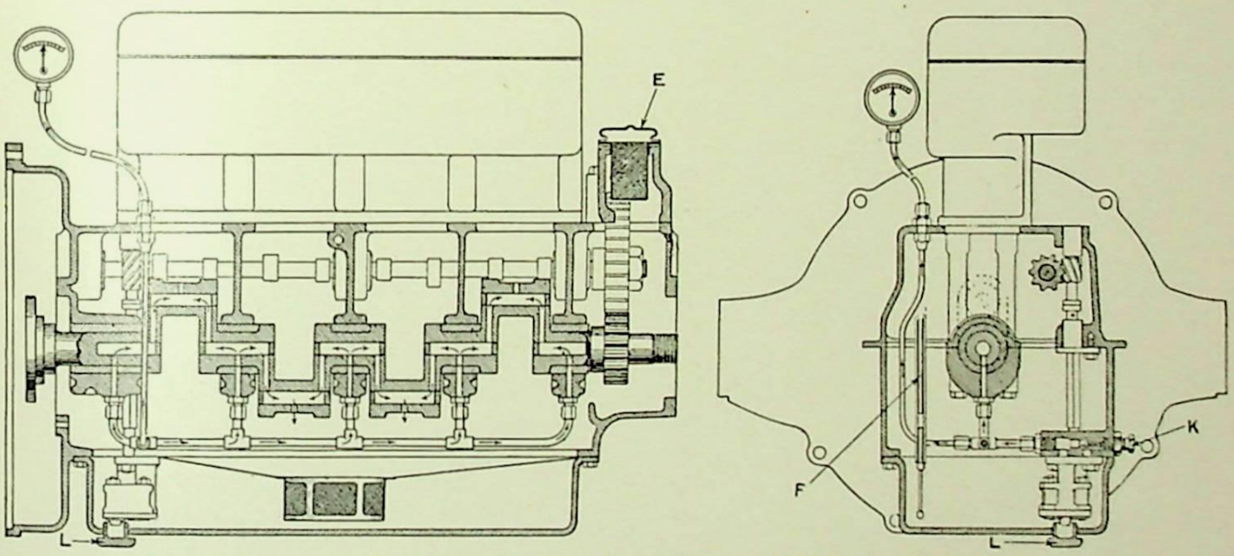


Fig. 183.—The Austin 12 h.p. Engine Pressure-lubrication System.

In this system it is essential to have a reliable type of pump, and a good oil strainer to prevent solid matter passing into and scoring the bearings. If the main bearings become worn, the oil will simply run out of the sides and will not pass through the drilled crankshaft. Thus if an engine begins to wear in this manner, the lubrication system will tend to become ineffective and the wear will become of increasing seriousness.

The oil pressure used in this system is only a few pounds per sq. inch, so that a simple type of pump only is required.

Fig. 183 illustrates the 12 h.p. Austin pressure system, which is similar in principle to the one previously mentioned.

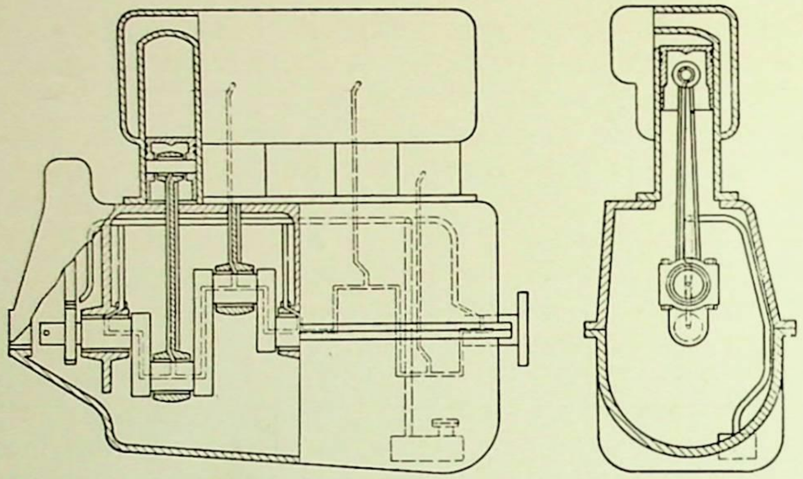


Fig. 184.—The Pressure System (*I.A.E.*).

The left-hand view shows very clearly both the drilled crankshaft and the direction of the oil flow. Oil is fed into the main crankshaft bearings at the two ends and three centre bearings, there being five separate main bearings and oil

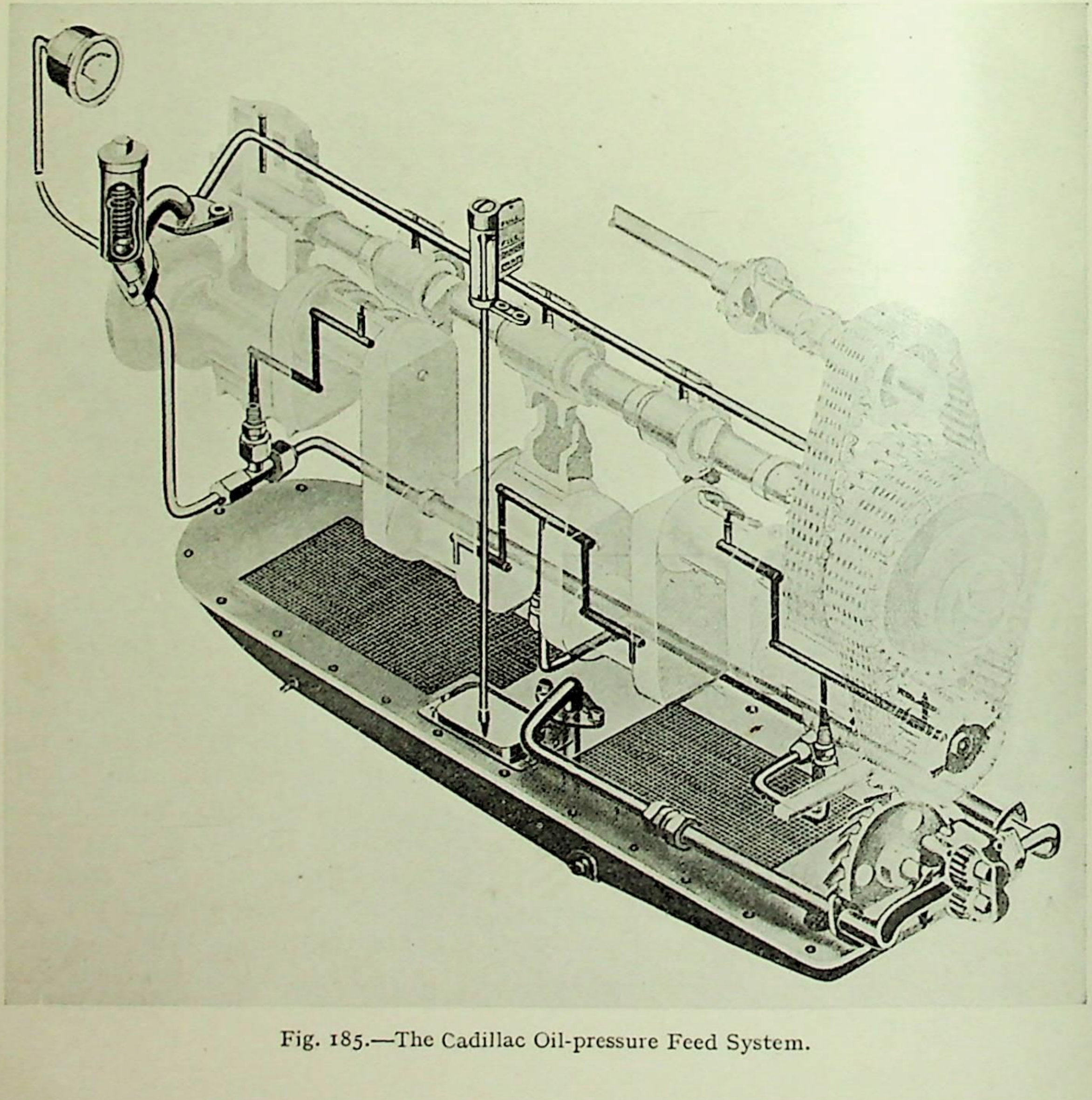


Fig. 185.—The Cadillac Oil-pressure Feed System.

inlets. The oil pressure normally required is 15 lb. per sq. inch, although the oil gauge registers a maximum of 40 lb. per sq. inch, since on chilly mornings when the engine is started the pressure may register 30 to 35 lb. per sq. inch until the engine warms up. The oil gauge is shown on the left-hand side, and leads from the lower supply pipe.

The oil filler E is of liberal capacity and has a gauze strainer. The dipper rod F is that previously shown in Fig. 175 for ascertaining the oil level.

In this system it is possible to adjust the oil pressure by regulating the position of the screw shown at K, a lock-nut being provided for fixing the adjustment when made. The vertical shaft for driving the submerged pump is driven by helical gearing from the left end of the camshaft as shown in Fig. 183.

THE FORCED-FEED SYSTEM

This is the most advanced system, and the one employed upon most automobile engines of high repute. It consists in forcing the oil under pressure to the main bearings, and thence through the drilled crankshaft to the big ends. Oil pipes also supply oil under pressure to the camshaft bearings, the timing gears, and frequently also copper pipes are securely fixed to the connecting rods to lead oil to the gudgeon pins. Occasionally this oil supply lubricates the pistons, although possibly the better way is to lead separate oil pipes to the base of each cylinder wall, feeding into an annular ring in the latter.

This system is employed on the Lanchester 40 h.p. and Vauxhall 14 h.p. engines, and in the former case the lubricating system can be followed out from Fig. 122. The gudgeon pins and the overhead camshaft are automatically lubricated in this case.

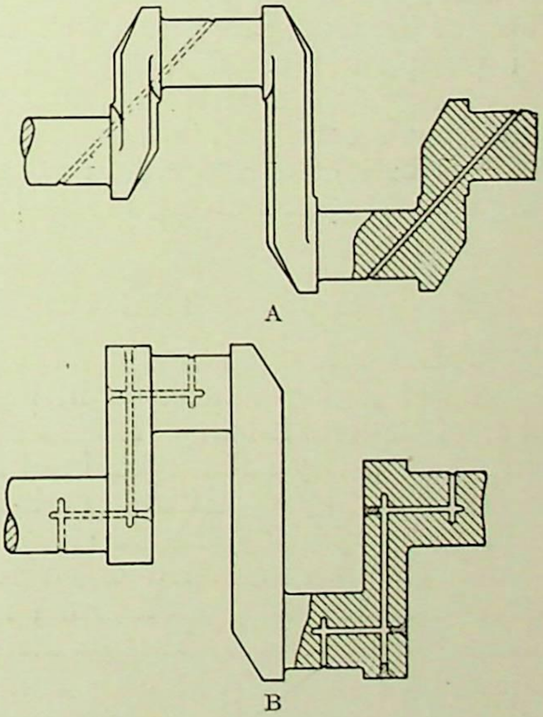


Fig. 186.

About $1\frac{1}{2}$ gallons of oil are carried in the sump, and the makers recommend replenishing every 250 to 400 miles.

The pressure system described is illustrated diagrammatically in Fig. 184, the oil pipes and oil channels in the crankshaft being indicated by the dotted lines in the left-hand illustration, and by the full line in the right-hand one. The Cadillac pressure system is shown in Fig. 185 in perspective view.

The oil pressure employed in this case varies from 15 to 40 lb. per sq. inch, and an oil-pressure gauge is invariably fitted on the dashboard with a warning division in red, below which the gauge needle must not fall when the engine is running.

It has been mentioned that the crankshaft is drilled when a pressure system is employed ; the alternative methods of drilling and plugging a crankshaft are shown in Fig. 186, of which the upper diagram method is usually adopted for short-throw cranks (i.e. short-stroke engines) and the lower diagram method for long-stroke engines.

THE DRY-SUMP METHOD

In this system, which is also a forced-feed one, no oil is kept in the crankcase proper, but there is usually a separate oil-chamber, or compartment, provided with cooling fins, either below the crankcase or on the dash. The oil from this chamber is pumped under pressure to the bearings, as in the preceding case, and thence drains down the sloping base of the crankcase into a well, from which a larger-capacity scavenger oil pump sucks the oil away and delivers it into the oil chamber. The advantage of this method is that the oil is cooled during its circulation and thus retains its lubricating properties better. Further, the lubrication is better controlled and no accumulation of oil can occur for the big ends to splash about, or dip into on an incline.

THE OIL-DISC METHOD

This simple method, which is employed occasionally upon the cheaper grades of engine, consists in fitting a large-diameter circular disc on one side of the crankshaft. This disc dips into an oil trough and is always therefore partly submerged in oil. As the disc rotates it flings the oil tangentially by centrifugal action, all around, and part of this oil is caught by specially shaped channels or troughs cast in the sides of the crank chamber, and led down to the main bearings and big-end oil troughs. Any surplus oil drains into the disc's trough, when it is again flung out as before. This method, it will be noted, does not employ a special oil pump.

OIL PUMPS

There are four principal types of oil pump employed on petrol engines, namely: (1) the Sliding-vane Eccentric Type ; (2) the Gear-wheel Type ; (3) the Centrifugal Type ; and (4) the Reciprocating Plunger Type.

The sliding-vane type is shown illustrated in Fig. 187. It consists of a rotating cylinder, the axis of which is set eccentrically to the circular casing. A pair of flat vanes having a central spring to keep them pressed against the casing can slide in the rotor. It will be seen that the rotation of the centre rotor anti-clockwise will cause the oil to be drawn into the crescent-shaped space under the influence of the suction created by the sliding vane, and the following vane will therefore scoop it forward into the delivery pipe. This type of pump is not, however, used very widely, partly on account of its small lift, and partly owing to the wear of the vanes and casing. It is usually submerged in the oil sump.

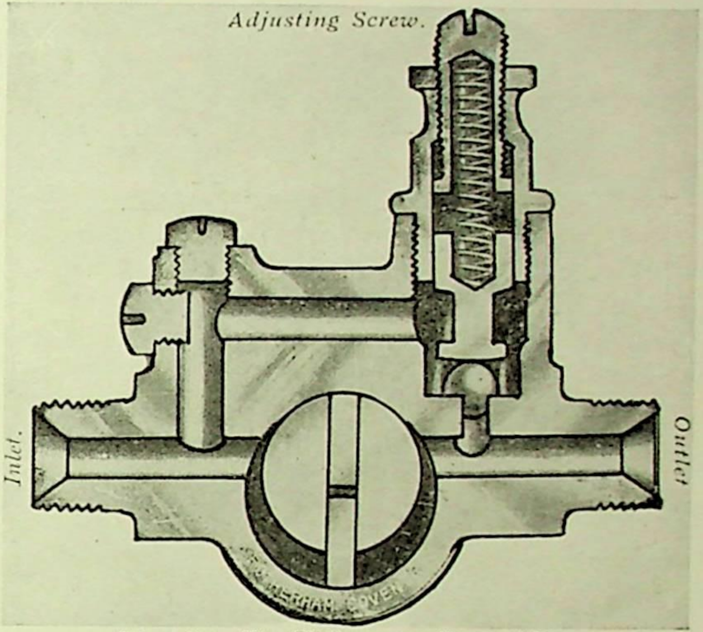


Fig. 187.—The Sliding-vane Type of Pump.

The gear-wheel type of pump is both simple to construct and will last for very long periods without losing its pumping efficiency ; for this reason it is widely employed on car engines. Moreover, there are no valves to get out of order in the gear-wheel pump. It consists of two or three toothed wheels meshing together in a casing, as shown in Fig. 188, so arranged that about one-half of the casing is in contact with each wheel. The width of the wheels and the inside of the casing are the same, except for a small working clearance. One of the wheels is direct-driven off a suitable part of the engine, and the inlet side of the pump is the side at which the teeth separate. The oil is carried around in the

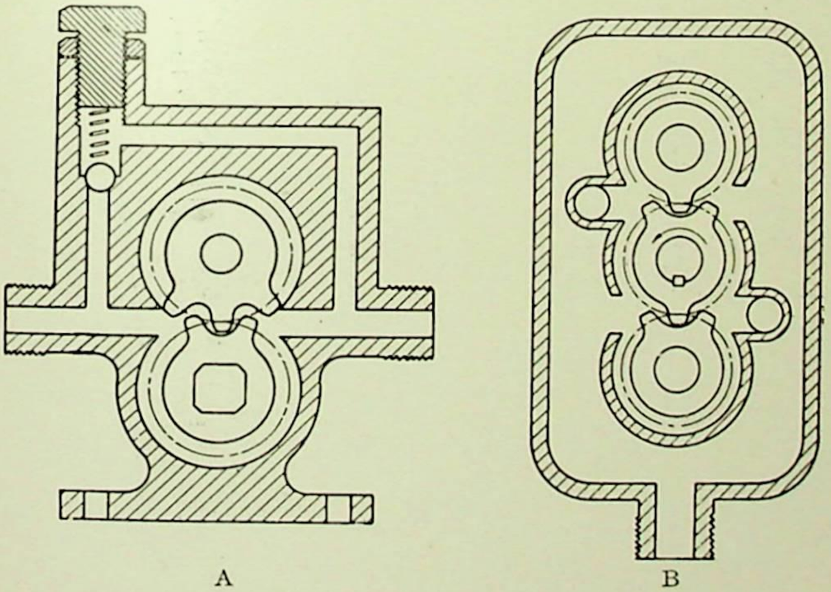
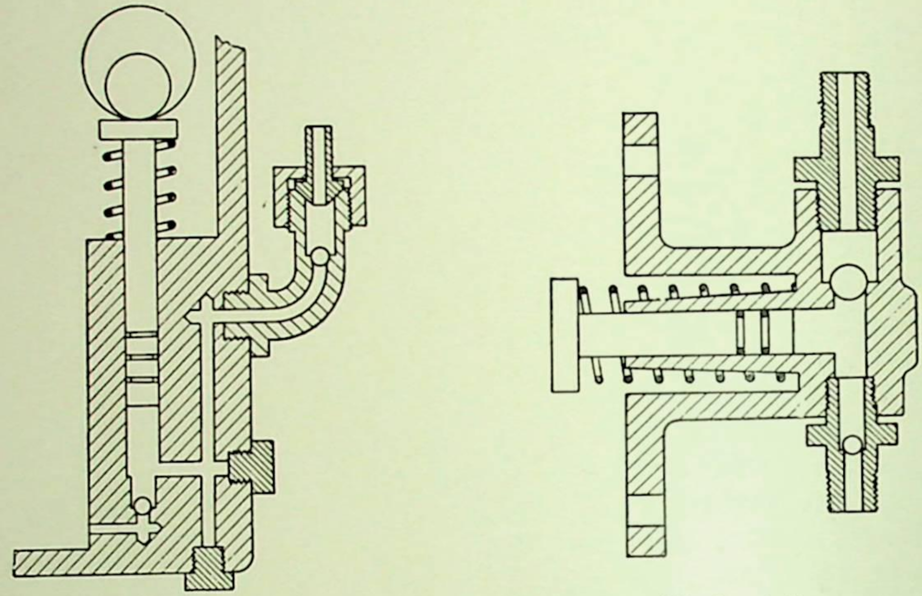
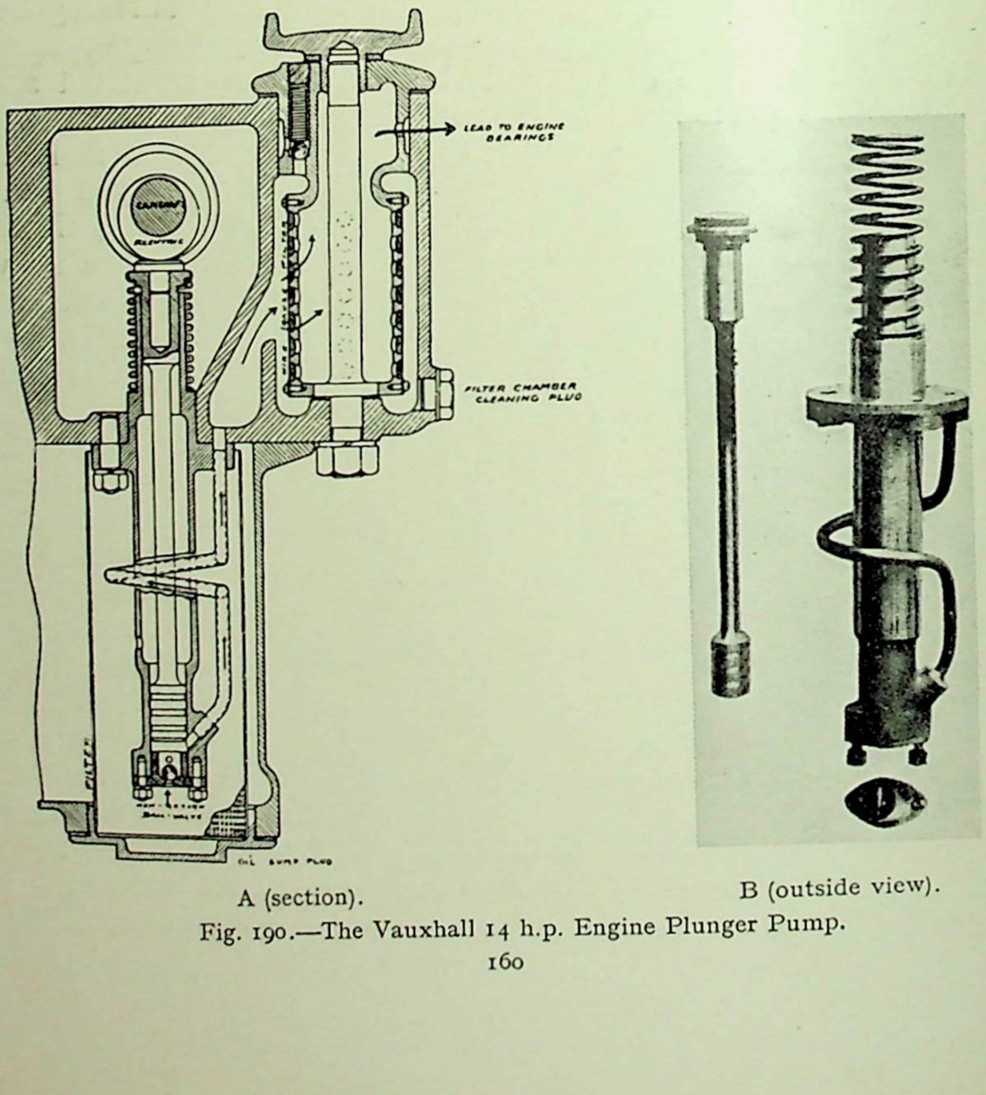


Fig. 188.—Showing Gear-wheel Type Pumps.



(Vertical) (Horizontal)

Fig. 189.—Showing Two Alternative Types of Plunger Pump.



A (section). B (outside view).

Fig. 190.—The Vauxhall 14 h.p. Engine Plunger Pump.

spaces between the teeth and is forced out again at the opposite side when the teeth meet again.

If D is the pitch diameter of the gear, P the diametrical pitch of the teeth, T the thickness of the gear (all in inches), and if N is the speed of the gear in revolutions per minute, then the theoretical delivery of the pump is given by:

$$D = \frac{6{\cdot}284\ T\ D\ N}{P} \text{ cubic inches per minute.}$$

The suction lift of this pump is very small, so that it should be placed as near the oil sump as possible.

The plunger type of pump is used upon the better classes of engine, as it is absolutely positive in action, and since it is the only type which will give a high pressure.

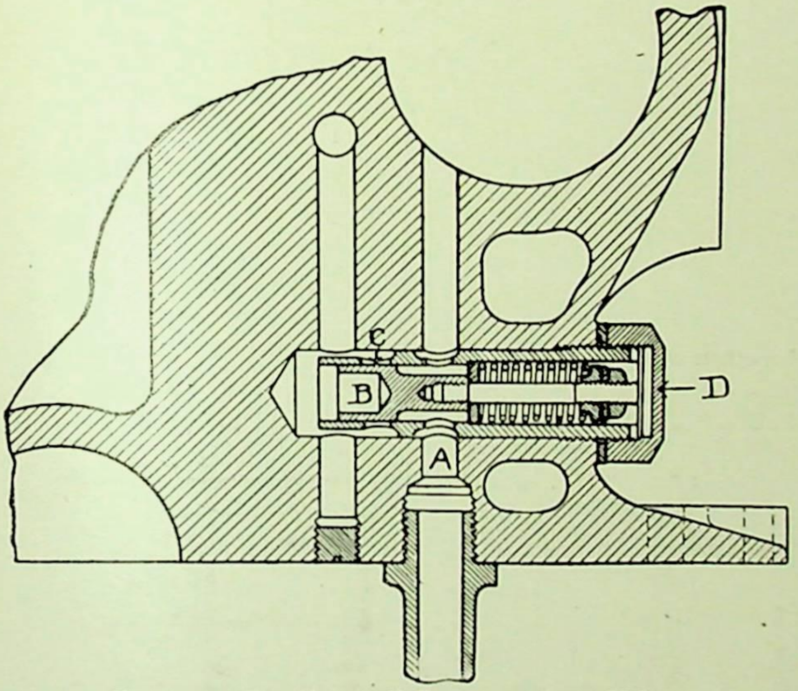


Fig. 191.—The Vauxhall Pressure-release Valve.

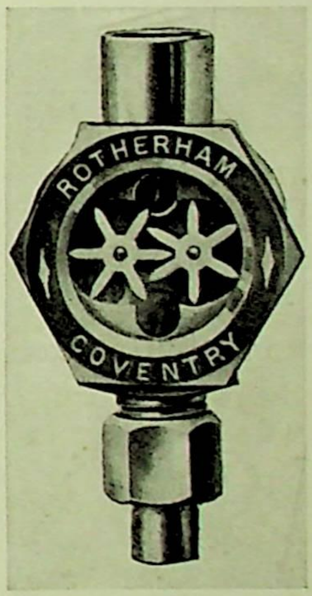


Fig. 191A.—A typical Oil-flow Indicator.

Fig. 189 illustrates two principal types of this pump. It will be seen that there is a reciprocating plunger which is forced in one direction (i.e. downwards) by means of an eccentric or cam, and is returned upwards by means of the spring shown. There is a ball non-return valve on the suction side, and another on the delivery side. The action of the pump is as follows: On the upward stroke a vacuum is created in the pump chamber, and oil is forced from the sump under atmospheric pressure influence past the non-return valve shown below, into the pump barrel. The downward stroke forces this oil through the delivery non-return valve above, since it is prevented from returning to the oil sump, and this action is continually repeated.

Fig. 190 shows the detailed construction of this type of pump in the case of the Vauxhall 14 h.p. engine. Here, the pump is shown enclosed at the base in a gauze filter for straining the oil of impurities, and the non-return ball valve is also shown in the base of the pump; the copper delivery pipe shown leads the oil to another chamber containing another gauze oil filter, whence the oil is forced to the bearings. It will be observed that the oil filters are both provided with means for readily detaching them for cleaning purposes.

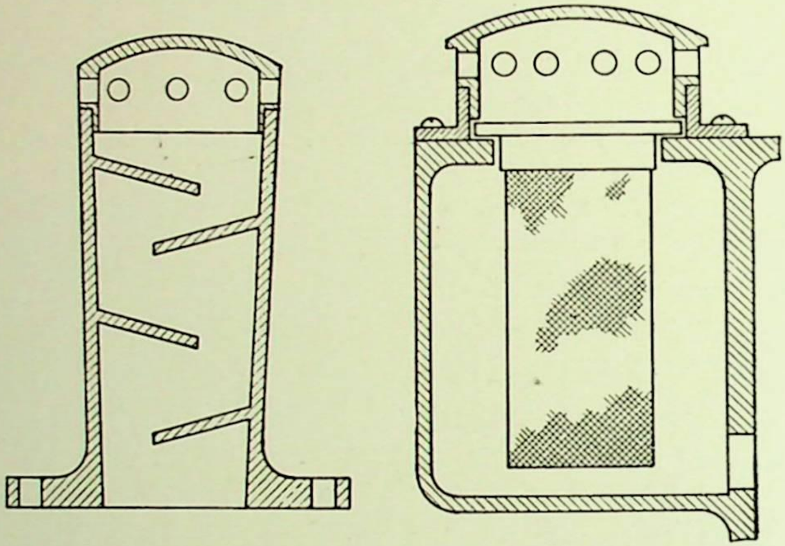

A B

Fig. 192.—Crankcase Breathers and Oil Fillers.

PRESSURE-RELIEF VALVE

It is usual to fit a relief valve in order to allow any excess pressure to find an outlet, and to return any oil

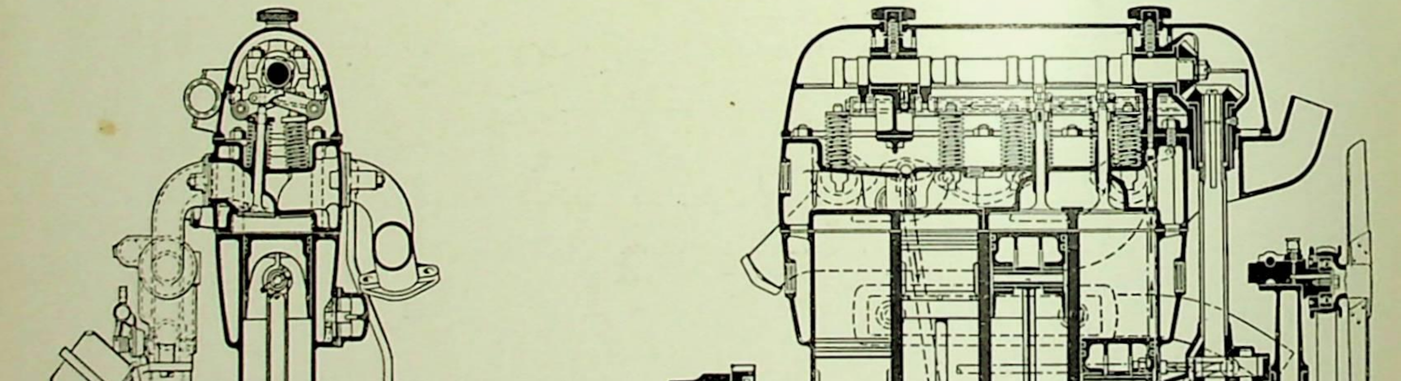


Fig. 193.—The Wolseley 10 h.p. Engine, showing Lubrication System (Splash-pressure).

surplus to requirements to the oil sump.[1] The Vauxhall engine relief valve is shown in section in Fig. 191. The oil under pressure enters the admission duct A, and bears against a spring-controlled piston B. When the oil pressure overcomes that of the spring, the piston will be forced along the barrel of the valve until the release duct C is uncovered, when it allows the surplus oil to be returned to the oil sump, after lubricating the timing wheels in its path.

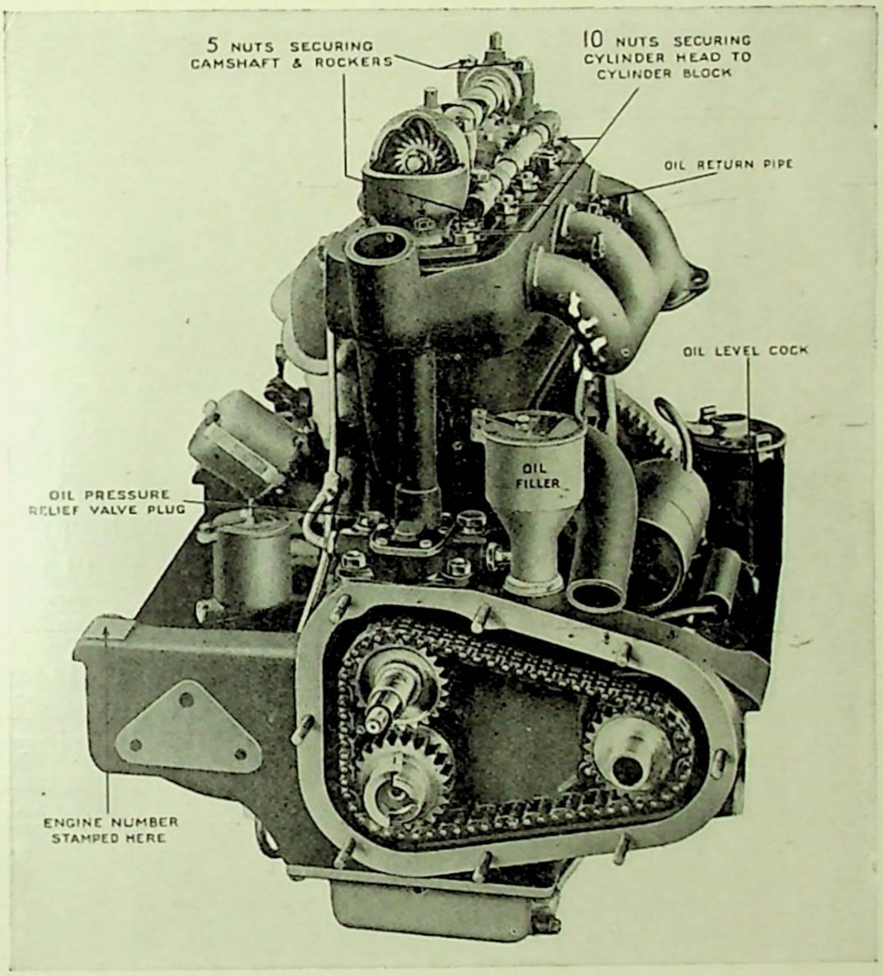

5 NUTS SECURING CAMSHAFT & ROCKERS
10 NUTS SECURING CYLINDER HEAD TO CYLINDER BLOCK
OIL RETURN PIPE
OIL LEVEL COCK
OIL FILLER
OIL PRESSURE RELIEF VALVE PLUG
ENGINE NUMBER STAMPED HERE

Fig. 194.—The Wolseley 10 h.p. Engine, showing Lubrication Parts.

OIL INDICATORS

Since the whole of the lubrication system is usually enclosed, it would be difficult for the ordinary person to ascertain whether the system was working properly, were it not for the fact that some form of oil indicator is usually fitted. The simplest form is an oil cock, or tap, introduced on the delivery side of the pump; if oil flows out when opened with the engine working, this indicates that the pump is functioning.

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[1] An example is shown also in Fig. 187, the ball valve on the right being a pressure-relief valve.

In the non-pressure splash systems it is usual to fit a dashboard type of indicator, usually with a glass front and a drip pipe through which the oil can be seen dripping when the system is working correctly. Another system employs a rotating wheel in a casing provided with a glass front ; oil impinges on the vanes of the wheel and rotates it (Fig. 191A).

In the pressure systems the pressure gauge is the best type of indicator, and is usually connected to the delivery side of the pump between it and the main-bearing supplies. Any choking of the oil passages is not shown by such a gauge, but the relief valve releases any abnormal pressures thus caused. A typical arrangement of pressure gauge is that shown in Fig. 181.

OIL FILLERS

Fig. 192 (A and B) illustrate two alternative designs of oil filler. In Fig. 192, A, it will be seen that there is a series of traps, the purpose of which is to prevent any oil being forced out owing to the displacement of air in the crankcase. This fitting is termed a *breather*, and it is also used as a filling tube, although no outlet is provided for the air to escape during the filling process. In the case of the design shown in Fig. 192, B, the breather is more effective as a filler, and it also incorporates an oil strainer.

In this connection it should be emphasised that it is extremely important to keep the lubricating oil very clean, as any solid matter would very quickly block up the oil ducts or score the bearing surfaces.

The external arrangement of the lubrication-system parts requiring attention by the driver of the car is well illustrated in the case of the 10 h.p. Wolseley engine shown in Fig. 194.

CHAPTER V

RADIATORS AND COOLING SYSTEMS

It has been mentioned in Chapter II that in an ordinary petrol engine the cooling system deals with about 20 to 28 per cent. of the heat energy of the fuel. As this heat energy is being continuously generated during the period in which the engine is running, it is evident that some means must be provided to carry off the surplus heat. In the case of air-cooled engines, radiating fins of large area are cast on the cylinder barrel and combustion head, and the heat conducted along these fins is radiated and swept off by a cooling current of air.

The amount of heat carried away will depend upon the length and section of the fins, the metal of which they are made, and the speed with which the cooling air rushes past.

Long fins, with thin edges, tapering in section from the cylinder barrel, or head, are the best, and although cast iron is convenient for air-cooled cylinders, aluminium is considerably better since it has a heat conductivity about 2¼ times that of cast iron, whilst copper is better still ; its heat conductivity being 4½ times that of iron.

For this reason air-cooled engines are sometimes made with an outer ribbed aluminium-alloy cylinder casting, in which a steel or cast-iron liner is forced, for the piston to work in, since aluminium itself is a bad wearing metal on account of its softness.

The copper-finned Chevrolet air-cooled engine has been referred to already (vide p. 134), and there is little doubt that this method has much to recommend it.

It is usual to make the radiating fins deepest around the cylinder head, where the greatest amount of heat is developed, and to taper them down, in length, along the cylinder barrel towards the crankcase. Around the exhaust valve the fins are made deep, and in the best practice the exhaust side is placed so as to receive the full force of the cooling draught first. The cylinder section shown in Fig. 21 illustrates the disposition of the fins in the case of a steel cylinder fitted with an aluminium-alloy head, in which bronze or cast-iron valve seats are cast or pressed.

The maximum permissible temperature of an air-cooled cylinder is about 300° C., the usual value being from 180° to 250° C. The cooling surface allowed for aluminium is about 0·5 sq. foot per b.h.p. of engine under these conditions, so that a 5 h.p. air-cooled engine would have about $2\frac{1}{2}$ sq. feet of area. For cast iron about 2 to $2\frac{1}{2}$ times this area must be allowed, to make up for its inferior heat conductivity. It is usual in the larger sizes of air-cooled engines to provide a fan of the pressure or suction type to force the cooling air past the fins, and in this case the axial type of radiating fin lends itself to better cooling than the usual motor-cycle type. In the case of a certain four-cylinder car engine of the six-cylinder 30 h.p. type, the bore and stroke were 92 mm. and 101·6 mm. respectively, and each cylinder was provided with a series of 56 steel fins parallel to the cylinder axis. Each fin was 0·064 inch thick, 0·785 inch deep, and 8·0 inch long. The cooling fan drew air past these fins at a speed of 18 m.p.h. when the engine was running at 1,200 r.p.m. It is estimated that the power absorbed by an efficient cooling fan works out at about 1 h.p. for every 15 to 20 h.p. given by the engine. The proportions of the cooling fins are given by Lanchester as follows:

Length of fin = 1·6 diameter of cylinder.
Spacing or pitch of fins = 0·03 diameter of cylinder.
Height of fins = 0·32 diameter of cylinder.

In the case of two-stroke engines about 60 to 70 per cent. more fin area must be allowed.

WATER-COOLED ENGINES

The water-cooled engine, as it is termed, is actually air-cooled, although indirectly so. The water in the cooling jackets has to be cooled, otherwise it would boil away, and serious damage would occur when the water level had sunk below the combustion head.

For this reason it is necessary to provide another cooling device, namely the radiator, to get rid of this surplus heat and to maintain the cylinder always at a temperature which must not reach the boiling point of water.

The water from the jackets surrounding the cylinders and valves is circulated either naturally (i.e. on the thermo-syphon principle) or "mechanically" (i.e. by circulating pump) through the jackets and through the core spaces of the radiator. Cooling air is induced to flow through the open spaces in the radiator, due both to the forward motion of the car and to the suction of a fan placed behind the radiator, inside the engine bonnet.

The two methods of cooling mentioned will now be described a little more fully.

The principle of the thermo-syphon system is simply the well-known water convection current property, which depends upon the fact that hot water is lighter, volume for volume, than cold water, and therefore, if water is heated, the hot water flows upwards and the cold water downwards, so that if there are no obstructions a natural circulation occurs.

Applied to engine cooling, as shown in Fig. 195, A, the hot water from the cylinders ascends along the water pipe or pipes leading from the top of the cylinders to near the top of the radiator, and descends through the radiator, being cooled in the process; the coolest water then flows from the bottom of the radiator up the intake water pipes and into the jackets from the bottom.

In the illustration shown, the lower pipe is horizontal, since it is the coolest water which must flow unobstructed into the jackets. It is therefore an essential principle in thermo-syphon cooling that the bottoms of the radiator and engine water jackets should be as nearly as possible on the same horizontal level.

Unfortunately, it is the exception rather than the rule (sometimes for constructional reasons) to follow this principle; a number of cars have their radiators fitted as shown in Fig. 195, B, and in which the cooling is by no means so effective as in Fig. 195, A. If the cylinder head is too near the top of the radiator, the water will endeavour to circulate in the wrong direction, i.e. it will oppose the proper circulation, and as a result the water will tend to remain stationary in the jackets and to boil there.

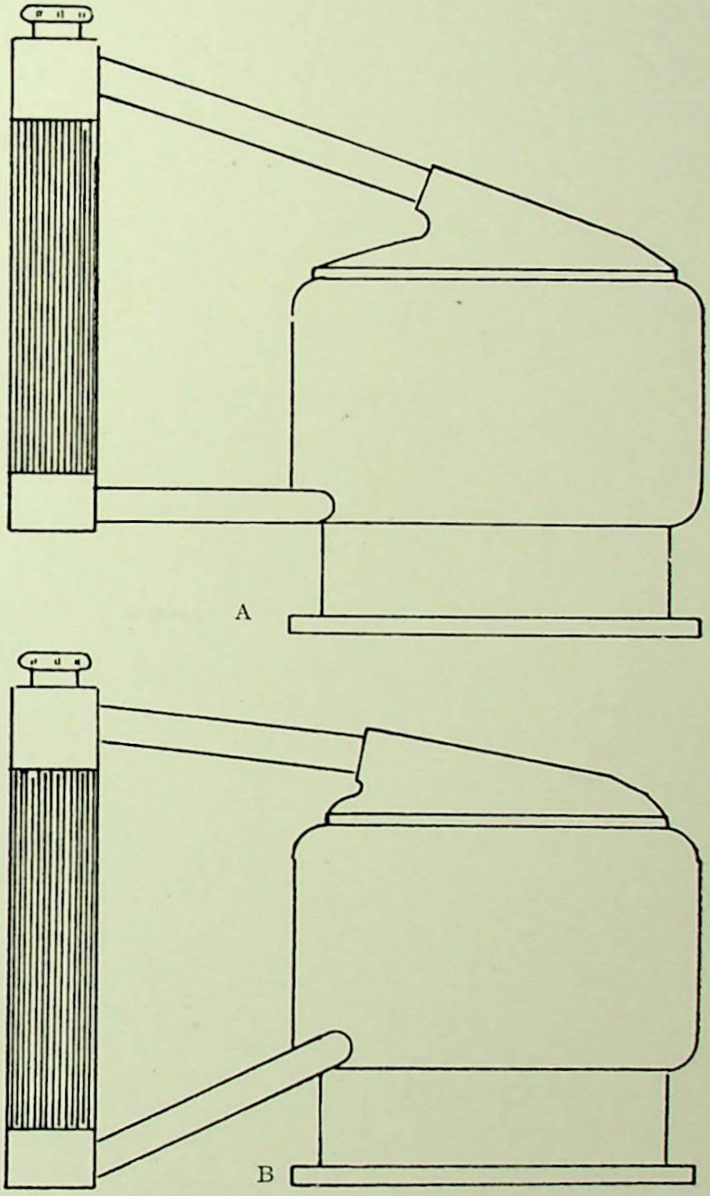

Fig. 195.

Fig. 196 shows the cooling system of the water-cooled Chevrolet car[1] engine. The radiator is provided with a lower water manifold, which is connected to the lower intake on the water jacket by means of rubber hose pipe, reinforced with canvas. The outlet from the cylinder jacket which conveys the hottest water is also connected in a similar manner to a pipe near the top of the radiator. Near to this pipe a smaller one can be seen leading from the top of the radiator down to the bottom of the radiator (outside). This pipe, which has an open end just inside the water-filling cap, is to carry away the steam, and, should the water become too hot and boil, to afford an escape for the water.

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[1] This system includes a pump, on the fanshaft.

The four-bladed cooling fan which draws air through the radiator can be seen in the illustration. It is driven from a Vee-pulley on the crankshaft by means of the round leather belt shown. The fabric-lined inner rim of the radiator is to afford a housing for the engine bonnet and to prevent the noise and rattle which would otherwise occur.

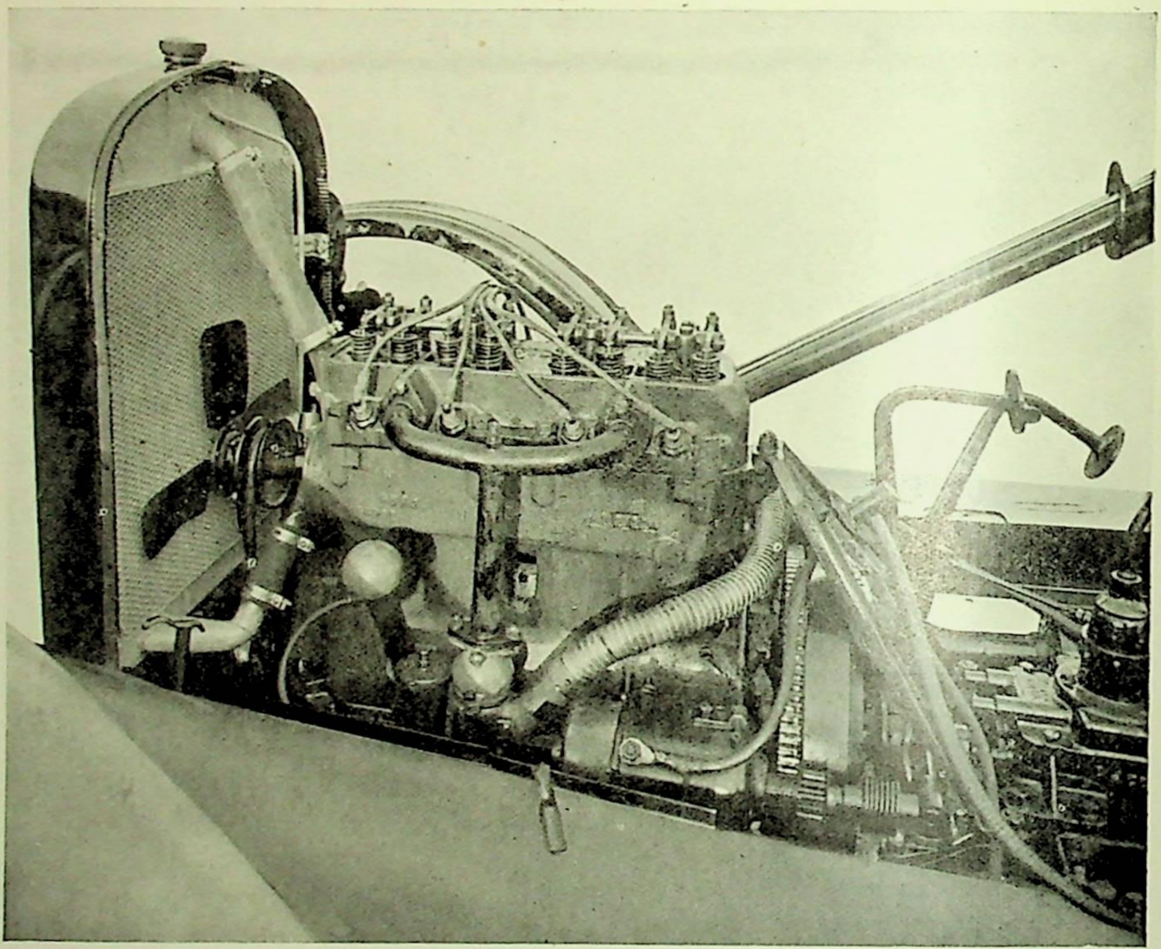


Fig. 196.—The Chevrolet Cooling System.

In the case of thermo-syphon cooling it is the usual practice to allow a radiator and water-jacket capacity of about 1 gallon for every 5 to 7 b.h.p. of the engine.

The fan for cooling air induction is usually mounted on a spigot bearing formed on the cylinder, and is provided with means for adjustment when the belt stretches. A typical arrangement is to mount the axis of rotation of the pulley eccentrically on the cylinder boss and to provide a locking device for the pulley shaft.

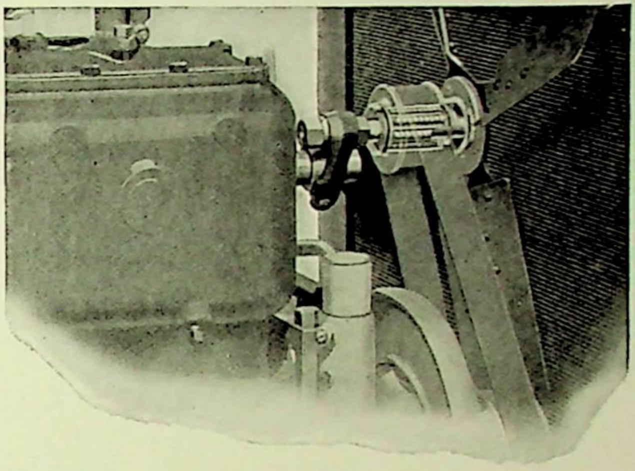


Fig. 197.—Showing Method of mounting Fan and adjusting Belt Tension.

Fig. 197 illustrates a swinging arm device for adjusting the (flat) belt; it also shows in part "phantom" view the Hyatt roller bearings in the fan-pulley boss, upon which the latter rotates.

Fig. 198 is a sectional view of a two-bladed metal fan pulley mounted on Hyatt helical roller bearings which run on a pin or overhung circular boss, mounted eccentrically to the fixed boss or pin which is clamped in a bracket (not shown) on the engine. The cooling fan is usually driven at from 1 to 2 times engine speed, and it will be evident that the speed of the cooling air will be proportional to the engine speed, or rather will increase with the engine speed.

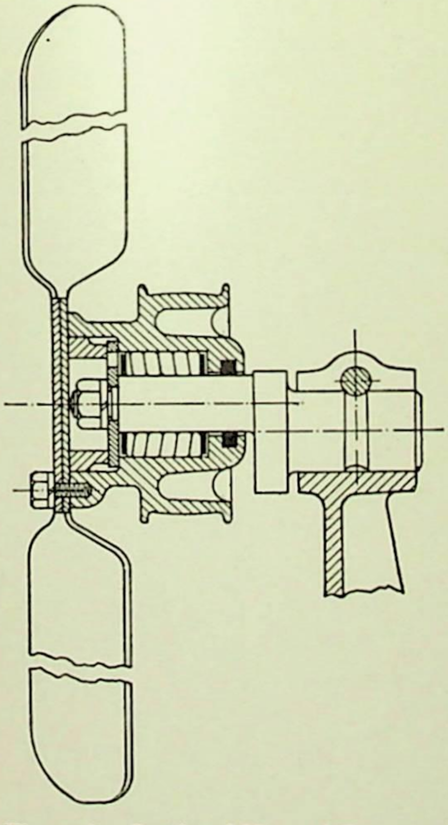


Fig. 198.—Sectional View of Fan, with Hyatt Roller Bearings.

PUMP CIRCULATION

With thermo-syphon cooling, larger water-jacket space and larger radiators are necessary, since the rate of circulation is very low. For large car engines very large radiators would be necessary, so that it is now usual to employ water-circulating pumps to speed up the circulation and thus to keep the sizes of radiators and water jackets down.

The pump employed is of the centrifugal type, the water entering near the centre of the rotating vanes of the pump and being ejected at the periphery; a small pump of a few inches' diameter can deal with a very large quantity of water in this manner.

The construction of a centrifugal pump is well illustrated in the example of the 12 h.p. Austin engine shown dissected in Fig. 199. The impeller, or rotor, is shown to the left of the part marked

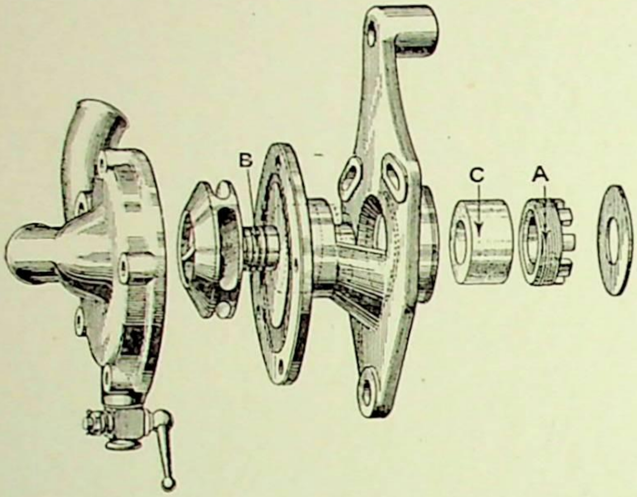


Fig. 199.—The Components of the 12 h.p. Austin Water Pump.

B; the pump spindle works in a gland which is provided with a sleeve C tightened up by means of the nut A. Between the sleeve C and the pump casing is a space which is filled with asbestos string soaked in tallow, to make a water-tight joint for the rotating pump spindle; a tap D is provided for draining the pump on frosty nights. The pump is situated between the radiator and the lower water inlet to the jackets and is usually driven from the secondary shaft which drives the magneto. Fig. 200 shows the arrangement of the pump, fan, and magneto in the case of the 14 h.p. four-cylinder Wolseley engine, and this illustration also indicates the various points which require attention; in the present considerations

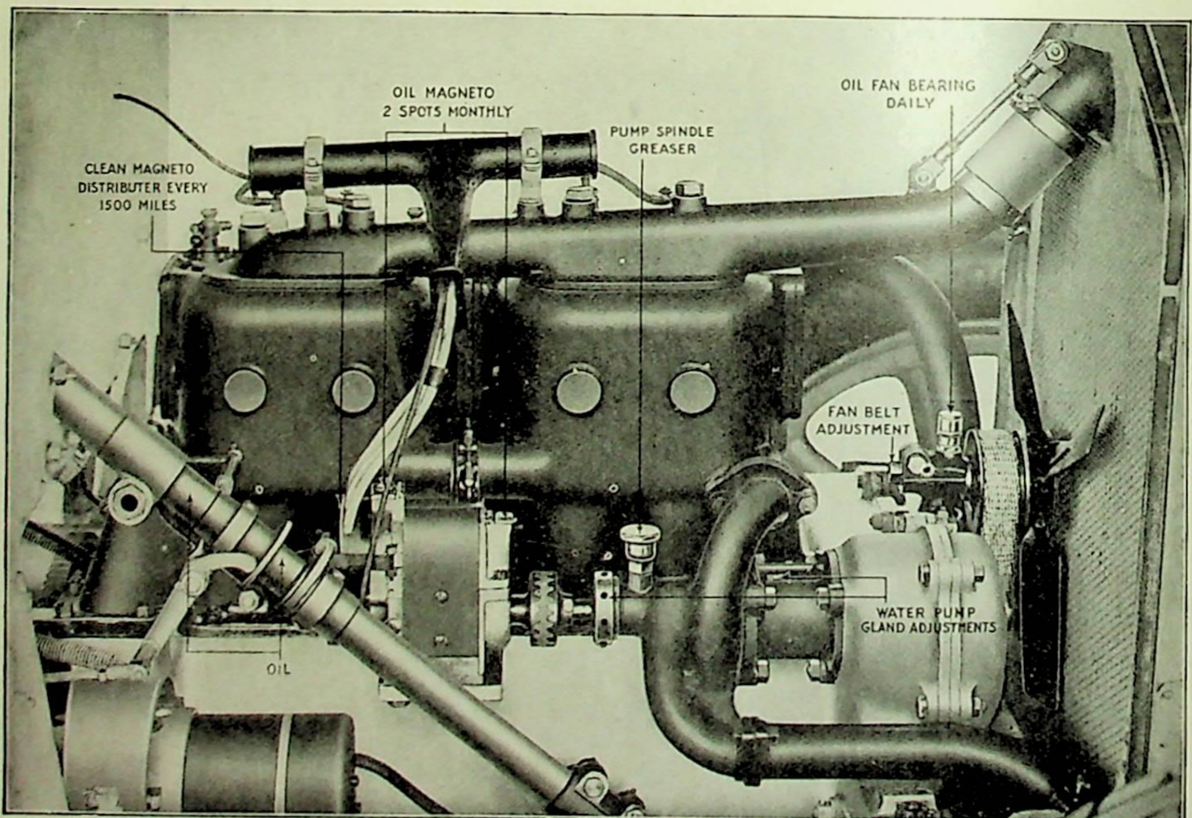


Fig. 200.—The 14 h.p. Wolseley Engine.

it is the fan-belt adjustment, the fan-bearing lubrication, the circulating pump spindle (grease) lubrication, and the pump gland adjustment to which periodical attention must be given.

The fan bearing requires oiling before every long run, and the pump spindle greasers also require frequent screwing down by small amounts.

In certain designs a compromise is arranged between the thermo-syphon and pump-circulation systems by fitting what is known as an *impeller,* that is, a small type of rotating fan placed in a circular casing situated between the radiator base and the cylinder jackets. This impeller accentuates the natural flow of water caused by thermo-syphonic action, but does not interfere with it. This impeller serves to force the cool water from the bottom of the radiator into the cylinder jacket. It is usually arranged on the same shaft as the fan pulley, and behind the fan (on the cylinder side). An excellent design of impeller is that fitted to the 15·9 Humber engine, and which is illustrated in section in Fig. 201. There the fan and impeller form one unit, which is secured to the top of the timing cover by four studs. It is driven by a Whittle (leather-steel link) belt from a pulley on the magneto shaft. The pulley is made adjustable to enable any slack in the belt to be taken up. The fan is mounted upon two ball races which fit on a shaft, which passes through into the main water circuit of the engine. Attached to the end of the shaft is a four-bladed impeller. To prevent water from leaking along the shaft to the ball races a spring-loaded gland is provided, which can be repacked when necessary by taking off the fan and removing the nut.

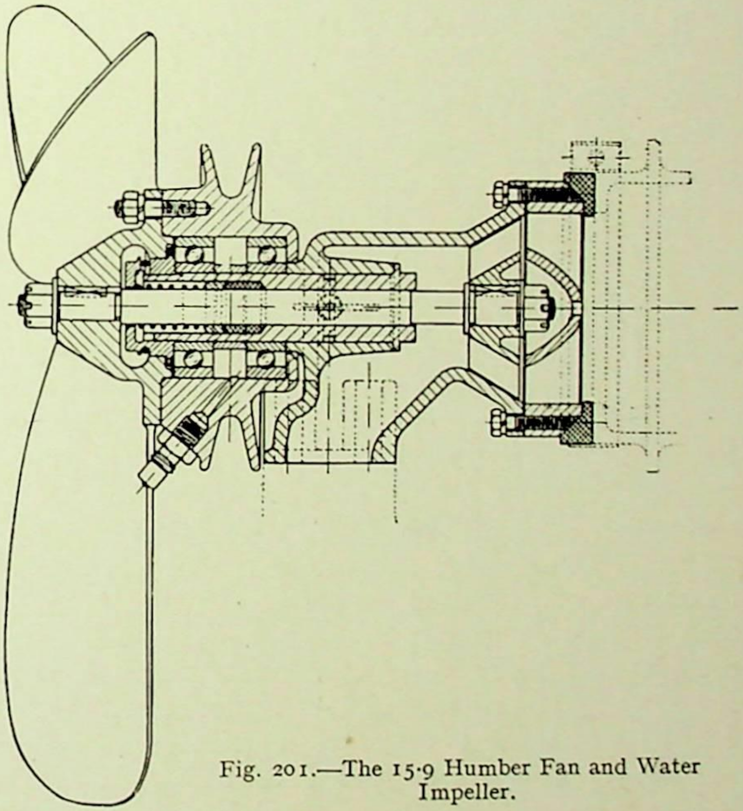


Fig. 201.—The 15·9 Humber Fan and Water Impeller.

QUANTITY OF WATER FOR COOLING

The amount of water required for cooling can also be computed from the rational formula :

$$Q = \frac{H}{t_1 - t_2},$$

where Q is the quantity of water in lb. (1 gallon = 10 lb.), H the number of heat units (B.T.U.) to be eliminated per minute by the cooling water, and t_1 and t_2 the inlet and outlet water temperatures Fahr. The number of heat units H can be taken as 2,520 per hour, or 42 per minute.

Thus in the case of a 20 h.p. engine, in which the difference between the inlet and outlet temperatures would be about 25° F., we have

$$Q = \frac{42 \times 20}{25} = 33{\cdot}6 \text{ lb. (i.e. } 3{\cdot}36 \text{ gallons).}$$

HAND-CONTROLLED AND AUTOMATIC TEMPERATURE REGULATORS

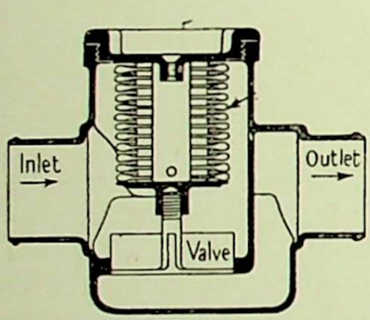


Fig. 202.—A Thermostat.

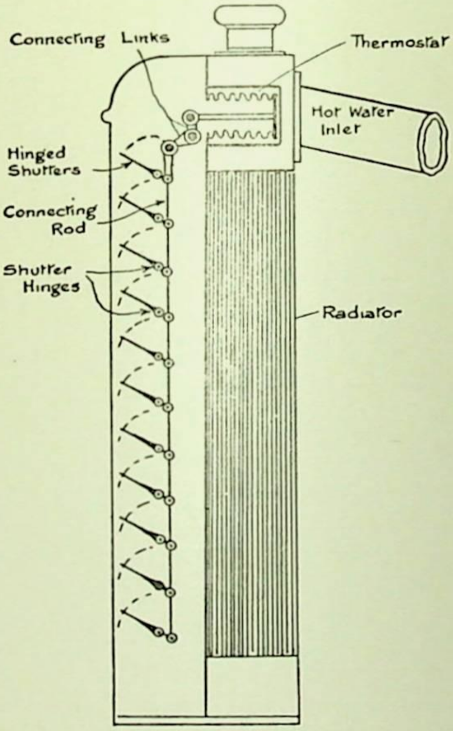


Fig. 202A.

It will be evident that if the radiator area is fixed, and also the quantity of cooling water carried is always the same, the engine will only be properly cooled by the methods just described under certain conditions, namely, level running. If the engine is called upon to climb long steep hills in hot weather, it will not be sufficiently cooled, whereas in very cold weather under light loads, such as coasting downhill, it will be over-cooled. No engine works efficiently under such conditions ; there is usually one particular small range of water temperatures at which it works best. In order to regulate or control the cooling-water temperature, radiator blinds or shutters of the pivoted type (resembling venetian blinds) are sometimes fitted to the radiator, and are automatically or hand-controlled in conjunction with a radiator thermometer. If the driver observes the temperature falling, he closes the shutters and therefore excludes air from the radiator tubes ; whereas if the temperature is observed to rise, he opens the shutters and admits more air through the tubes.

This method is a step in the right direction, but it requires the frequent attention of the driver. It has the advantage of being able to warm the engine up more quickly, since the car can be run with the shutters closed until it heats up to normal conditions.

It is now usual, in the case of the better-class cars, to employ an automatic means for controlling the temperature. The usual method consists in inserting a double poppet valve in the water system controlled by means of a thermostat. The latter device generally consists of a series of small, hollow, thin, copper discs in the form of a bellows, filled with a volatile liquid ; the effect of an increase in temperature is to cause the liquid to volatise and expand, thus increasing the length of the disc system. This change of length is utilised to open the poppet type of valve and to allow more water to flow through. This thermostatic control is fitted between the cylinder head and the radiator, so that when the engine is cold, the discs being in contraction hold the valves on their seating and prevent any water from the cylinder heads from reaching the radiator ; the radiator is thus cut off until the engine warms up, when the valve is opened by the expansion of the thermostat and water flows to the head of the radiator. It will also be evident that the thermostat will regulate the flow of water according to its temperature, and will, if correctly designed, keep the cooling water constant in temperature. The thermostat is occasionally used to control the radiator shutter as shown in Fig. 202A.

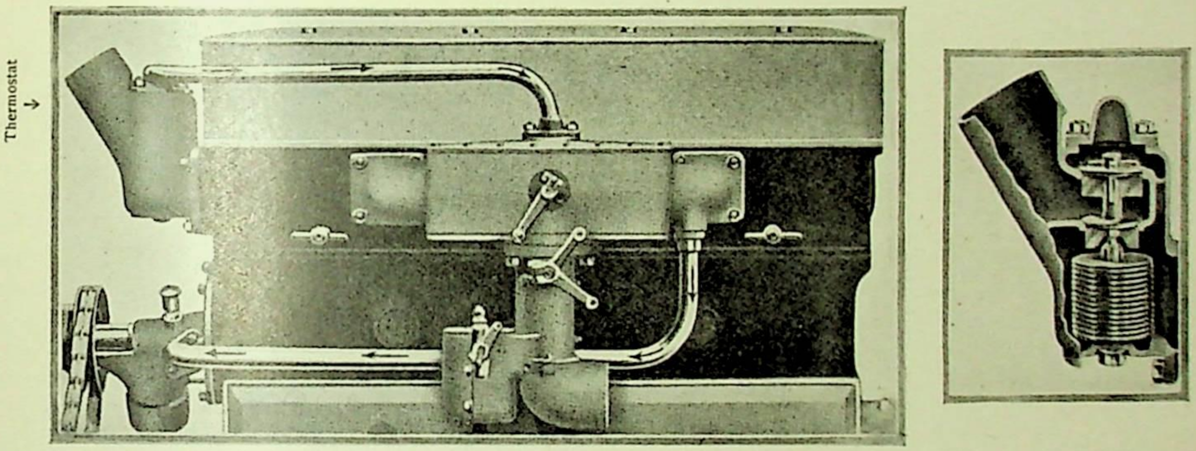


Fig. 203.—The Vauxhall Thermostat (right) and Installation (left).

An example of a thermostat is given in Fig. 203, which shows the construction of the thermostatic device fitted to the Vauxhall engines. Fig. 203 shows also the method of fitting the thermostat in the cylinder-head outlet pipe in the case of the 23–60 h.p. Vauxhall engine ; this illustration indicates also the by-pass for the hot water through the carburettor mixing chamber, as indicated by the arrows.

CONSTRUCTION OF RADIATORS

The radiator is designed so as to provide the maximum area of cooling surface for a given quantity of water, and with a series of free paths and passages for the air, so that the heat conducted from the cooling water shall be dissipated as freely as possible.

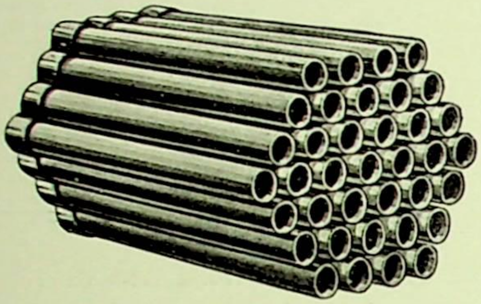
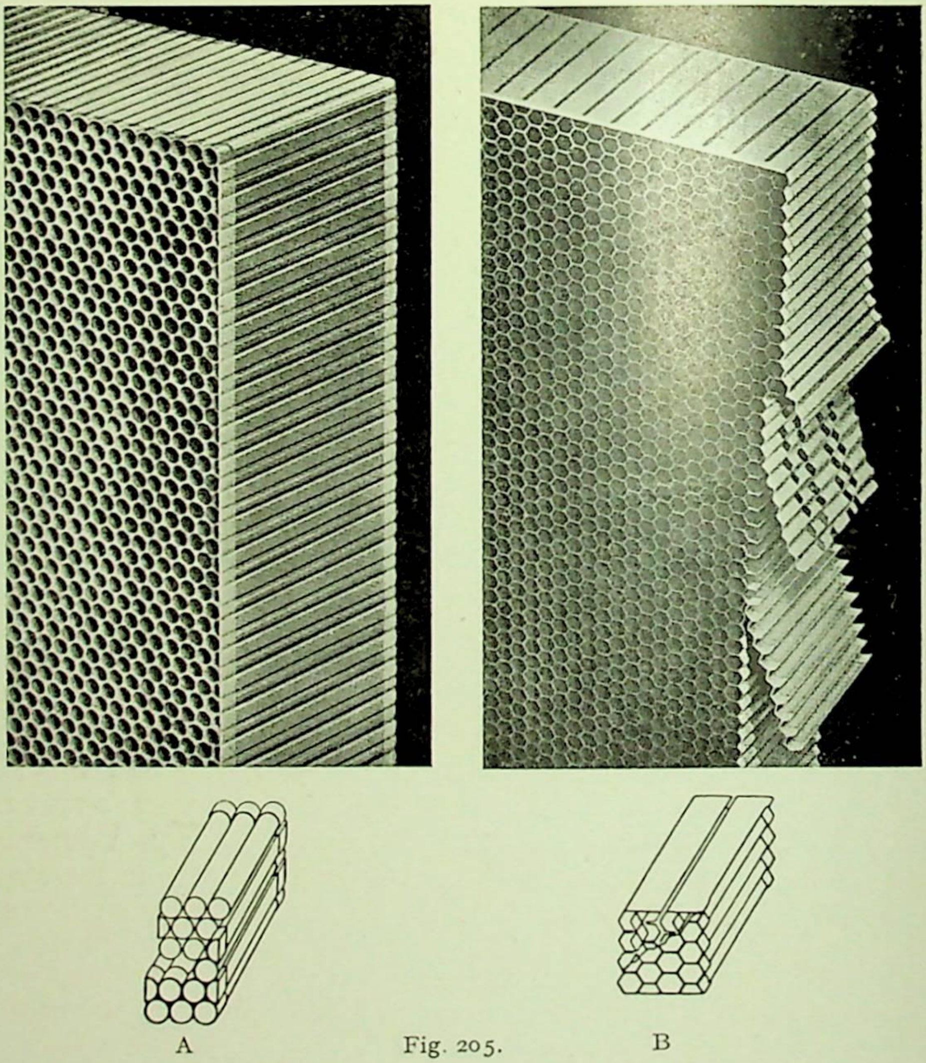


Fig. 204.

There are three principal methods of construction of the radiator system, namely: (1) the round honeycomb tube; (2) the film or corrugated-strip type; and (3) the gilled-tube type.

The method of construction of the honeycomb radiator, as carried out by Messrs. Serck, is shown in Fig. 204, a small nest of round tubes being shown joined together so as to provide air passages through the tubes and water spaces between them. Fig. 205, A, illustrates the Marston radiator, which is also made on the honeycomb system, some thousands of separate, very thin brass tubes, expanded at each end, being soldered together at the ends only. The tubes are usually only about $\frac{5}{1000}$ to $\frac{10}{1000}$ inch thick. It is usual to assemble the tubes in a frame of the desired finished shape, and to further expand the ends of the tubes so as to keep them in intimate contact, and then to immerse each of the two faces of the blocks in a bath of solder



A Fig. 205. B

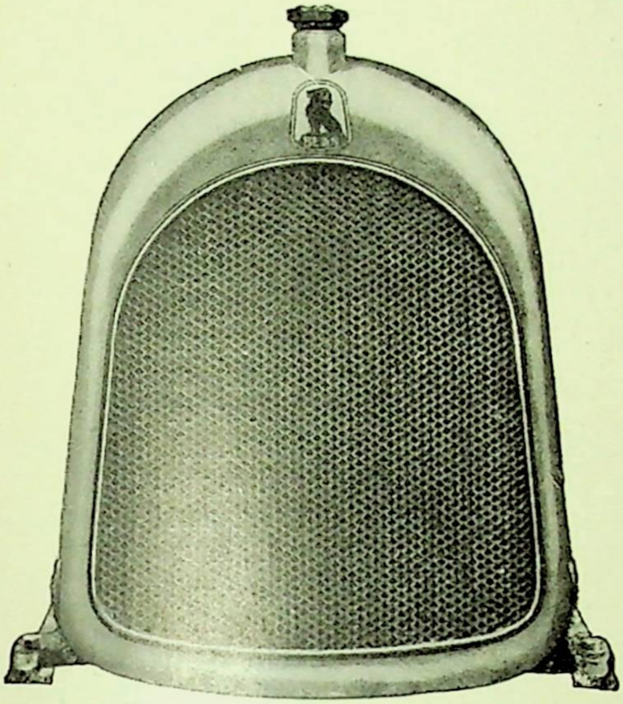


Fig. 206.—The Bean Radiator (Serck).

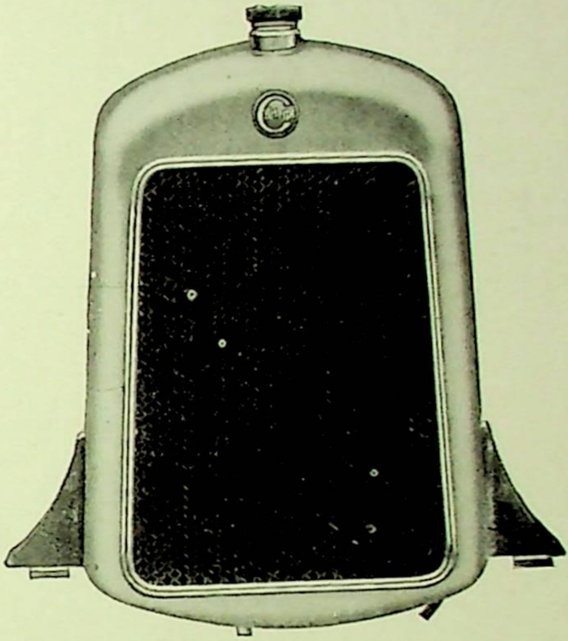


Fig. 207.—Calthorpe Radiator (Serck).

for a few seconds in order to fill all the spaces between the tubes at the ends.

In the second method the tubes extend right from the top to the bottom of the radiator. This type of tube is made up of two long strips of corrugated brass with the edges expanded, and when these are soldered together it gives the necessary waterway from the top to the bottom. The desired quantity of these tubes is then formed into one block and is inserted into the radiator case. Fig. 205, B, illustrates the method of construction of a strip type of radiator employed by Messrs. Marstons.

Some examples of finished radiators made by these methods by Messrs. Serck are shown in Figs. 206, 207, and 208.

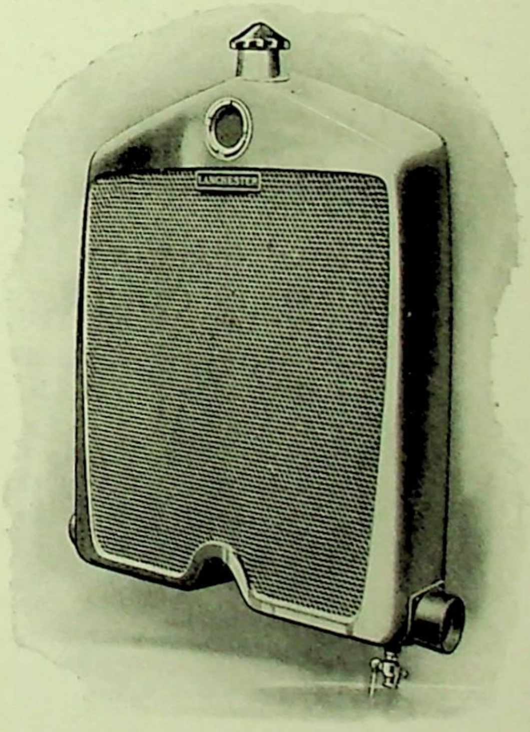


Fig. 208.—Lanchester Radiator with Water-level Window.

In the case of the Lanchester radiator, a glass window is inserted near the top in order to show the level of the water in the radiator; the usual method is to unscrew the filler cap—when one remembers to do so.

The radiator is generally made as one unit, namely, of a stamped radiator case, which is afterwards nickel plated (as a rule), and from which the centre portion is cut away, for the tube block to be inserted and soldered in position. Another method which is used on the Ford car, and in several other cases, is to solder the radiator block to the top and bottom tanks, and to slip the completed unit into another outer case, which is constructed simply for appearance purposes only.

The gilled-tube system of construction consists of an upper and a lower water tank connected by a large number of copper tubes, each provided with a series of gills or fins; these are usually corrugated and wound spirally around the tubes, which are of from $\frac{3}{16}$- to $\frac{1}{2}$-inch diameter. This system is not so efficient as the others just described, but is cheaper in construction. Its use is now confined practically to commercial vehicles.

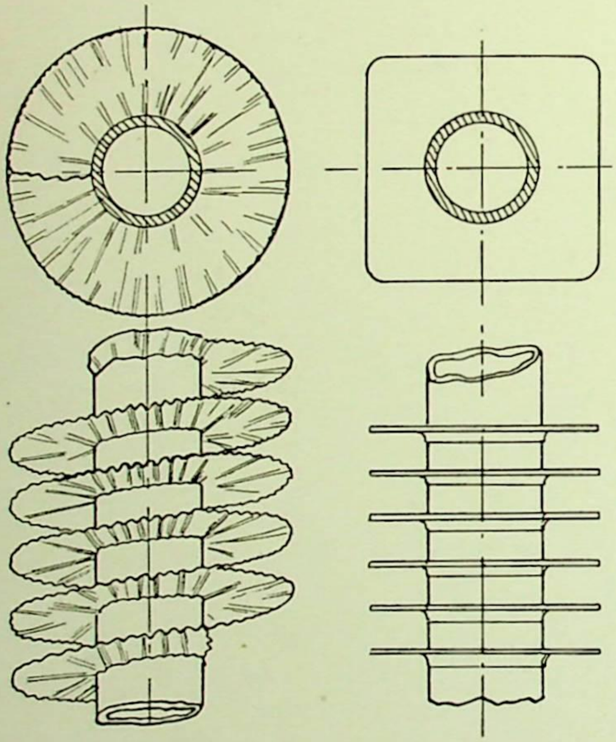


Fig. 209.—Gilled Tube Radiator Elements.

The metals used for the radiator cores, in the first two methods, are brass, copper, and nickel silver, the latter being employed where a white metallic finish is required. The average thickness of the metal employed for brass honeycomb radiators is about $\frac{7}{1000}$ inch, and the percentage of free area through which the air passes varies from 50 to 85 per cent. of the frontal area of the core.

The water tanks are usually made in 18- or 20-gauge sheet metal, and the size of the inlet and outlet pipes fittings varies from 1 inch, diameter (inside) for engines up to 10 h.p., up to 2 inches for 20 to 30 h.p. engines with pump circulation, and from 50 to 100 per cent. greater diameter for thermo-syphon.

It is usual to allow a total area of radiator cooling surface of from 2·3 to 3·0 sq. feet per b.h.p., and a core depth of from 2 to 4 inches.

The frontal area of a well-designed radiator as constructed on either of the first two methods described, and for a core depth of 2 to 3 inches, should be about 8 sq. inches per b.h.p. for motor cars, and about 12 sq. inches per b.h.p. for commercial vehicles using the pump system; for thermo-syphon cooling these areas must of course be made greater, an increase of 30 to 40 per cent. being usual.

THE FIXING OF RADIATORS

The radiator should be fixed to the chassis frame front cross-member in such a manner that any flexure of the frame will not stress the radiator ; otherwise the more or less delicate soldered joints of the water headers and thin radiator tubes will become strained and will show signs of leakage.

The common method is to bolt the radiator to the frame at the two lower sides of the radiator, by means of angle brackets secured to the radiator, as shown in Fig. 210 ; pieces of packing such as rubber or canvas strips are interposed in the joints so as to reduce vibration. The inlet and outlet pipes of the radiator are connected to the engine cylinder connections by means of rubber-hose pipes, the object being an easily detachable, water-tight, and vibration-proof connection in each case.

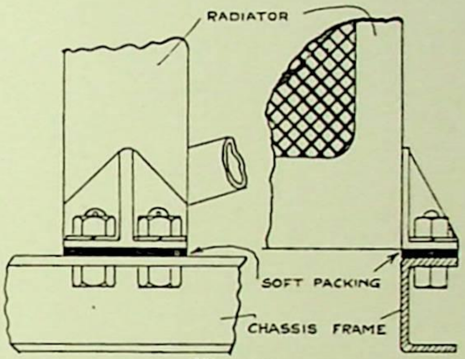


Fig. 210.

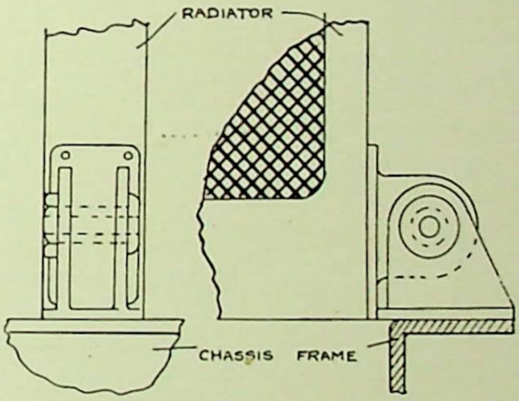


Fig. 211.

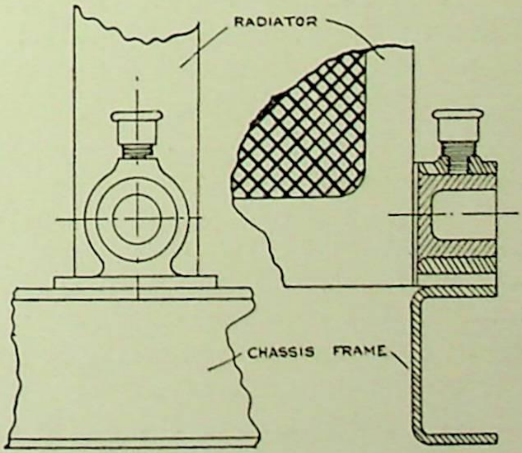


Fig. 212.

In some cases the radiator is mounted on horizontal trunnions which can rock in bearings fixed to the frame, as shown in Fig. 212. This method relieves the radiator of the stresses to which the frame may be subjected ; it is necessary to grease these trunnions periodically, since there is always some slight movement of rapid frequency. In another method, which is employed on the Aster six-cylinder engine car, the radiator is mounted on non-flexible three-point supports on the chassis frame. This method of "three-point" support is frequently employed for the engine and gear-box anchorage to the frame, and is intended to relieve them from frame-distortion stresses.

CARE OF THE RADIATOR SYSTEM

The passages between the cooling fins, through which the cooling water flows downwards in a series of fine streams, are very small in width and area, so that any obstructions therein will tend to reduce the water flow. For this reason it is essential to strain the water before filling the radiator and to use only "soft" water, such as rain-water. The effect of a "hard" water is to cause a deposit of lime or magnesia in the radiator tubes and passages, which will seriously affect the cooling efficiency in time. This deposit can be removed by filling the radiator with a strong boiling solution of washing soda in water, taking care that none of this solution is splashed over the paintwork

It is an advantage to have a radiator thermometer fixed in the top of the radiator, so that the temperature of the hottest portion of the cooling water can always be noted. These thermometers, of which the Boyce Motometer, Markt and Smith instruments are satisfactory examples, are not only useful for this purpose, but at once indicate when the water level is too low by showing a low reading.[1]

The fan belt should be properly tensioned, and the circulating pump gland packed and adjusted to enable the pump spindle to run without undue friction, but with no leakage of water.

In cold weather, when the temperature of the car is likely to be at or below zero, the water should be drained from the radiator through the plug or tap provided for this purpose, and care should be taken that the pump casing and carburettor jackets are also drained out as well.

It is possible, however, to use an "anti-freezing" solution in the radiator, which will not freeze at several degrees below zero. Glycerine is frequently used for this purpose. Thus a 10 per cent. solution in water freezes at 2° Fahr. below the freezing point of water, a 30 per cent. solution at 7° Fahr. below, and a 40 per cent. solution at 32° Fahr. below the freezing point of water. It is inadvisable to use more than a 20 per cent. solution of glycerine, however, as it affects the rubber connections detrimentally.

A 15 to 20 per cent. solution of alcohol will also lower the freezing point several degrees; whilst another useful mixture is one composed of 5 lb. of calcium chloride and a handful of washing soda in every gallon of water; this mixture should be well stirred and strained and any evaporation in the radiator made up for by addition of the same mixture, not by water alone.

In cold weather the radiator should be covered with a rug, or with one of the radiator muffs sold for the purpose, when the car is left standing for a time, or is being run up in the garage or open.

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[1] These instruments are described in Volume III of this work.

CHAPTER VI

FUELS AND CARBURATION

THERE are many liquid fuels suitable for internal-combustion engines, ranging from the heavy, tarry oils used in Diesel engines right up to the very light and highly volatile petrols employed in aircraft engines. Most of the fuels in use are obtained by distillation from coal tar or from crude petroleum, and consist of organic compounds containing hydrogen and carbon (with the addition, in some cases, of oxygen). The former compounds are termed hydrocarbons. The fuels employed in automobile engines are petrols of various grades, ranging in specific gravity from 0·68 to 0·75, benzole—a coal-tar derivative—paraffin of the well-known illuminating-oil grades, and alcohol.

The principal properties of a fuel with which we are here concerned are its density (i.e. its weight per gallon, say), heating value, volatising properties (or vapour pressure), flash points, and range of combustion, or air-fuel proportions.

In the following table the principal properties of the more important fuels are given, together with the recommended compression pressures for internal-combustion engine use.

Fuel.	Heating Value per gallon B.T.U's.	Weight per gallon (lb.)	Chemical Formula.	Air-Fuel for Best Combustion, (by weight).	Range of Compression Pressures, lb. per sq. inch.
Aviation Petrol	131,500	6·7	$C_6 H_{14}$	15·0	50 to 125
Commercial Petrol, Grade 1	133,500	7·1	Mixtures	14·8 to 15·3	50 to 120
Commercial Petrol, Grade 2	140,000	7·5	Mixtures	14·8 to 15·3	50 to 100
Commercial Benzole	144,000	8·6 to 8·9	$C_6 H_6$	13 to 13·5	50 to 145
Petrol-Benzole, 50/50 mixture	140,000	7·6 to 8·0	Mixture	14 to 14·5	50 to 135
Paraffin	160,000 to 175,000	7·9 to 8·3	Mixture	13 to 14	50 to 80
Alcohol, 100%	92,800	7·95	$C_2 H_6 O$	9 to 10	100 to 200
Alcohol, 90%	82,100	8·15	—	—	—
Alcohol, 80%	70,000	8·36	—	—	—

PETROL

Petrol is obtained by the distillation of crude petroleum, and consists of the light vapours which are volatised when the latter is heated to about 70° C. The lightest petrol, namely, the aviation grade, has a density of ·66 to ·68 and is extremely volatile. Petrol of the commercial grades varies from ·70 to ·78 in density, and is a mixture of the lighter hydrocarbon, hexane ($C_6 H_{14}$), with other petroleum spirits, such as heptane and pentane. On the average, a pound of petrol contains 0·84 lb. of carbon and 0·16 lb. of hydrogen, and has a heating value of about 19,000 to 20,000 British thermal units per lb., that is to say, the heat obtained by burning a pound of petrol would, if no heat were otherwise lost, raise the temperature of 190 to 200 lb. of water (i.e. 19 to 20 gallons) from 0° C. to 100° C.

It is the heating value of a fuel which is an important factor in its use in combustion engines.

Petrol vapour mixed with air forms an explosive mixture, the range over which it will explode varying from about 8 lb. of air to 1 lb. of petrol in the case of a rich mixture, to about 20 lb. of air to 1 lb. of petrol for the weakest mixture. The best mixture for perfect combustion is one of about 15 lb. of air to 1 lb. of petrol. Experiments show, however, that a rather richer mixture, namely, one of from 12 to 14 lb. of air to 1 lb. of petrol, gives the greatest power results in a petrol engine, whilst a rather weaker mixture, namely, from 16 to 18 lb. of [illegible] lb. of petrol, gives the best fuel economy.

The compressions which can be employed, with petrol as a fuel, vary from about 50 to 125 lb. per sq. inch, the higher compressions being used in high-speed and high-efficiency car engines.

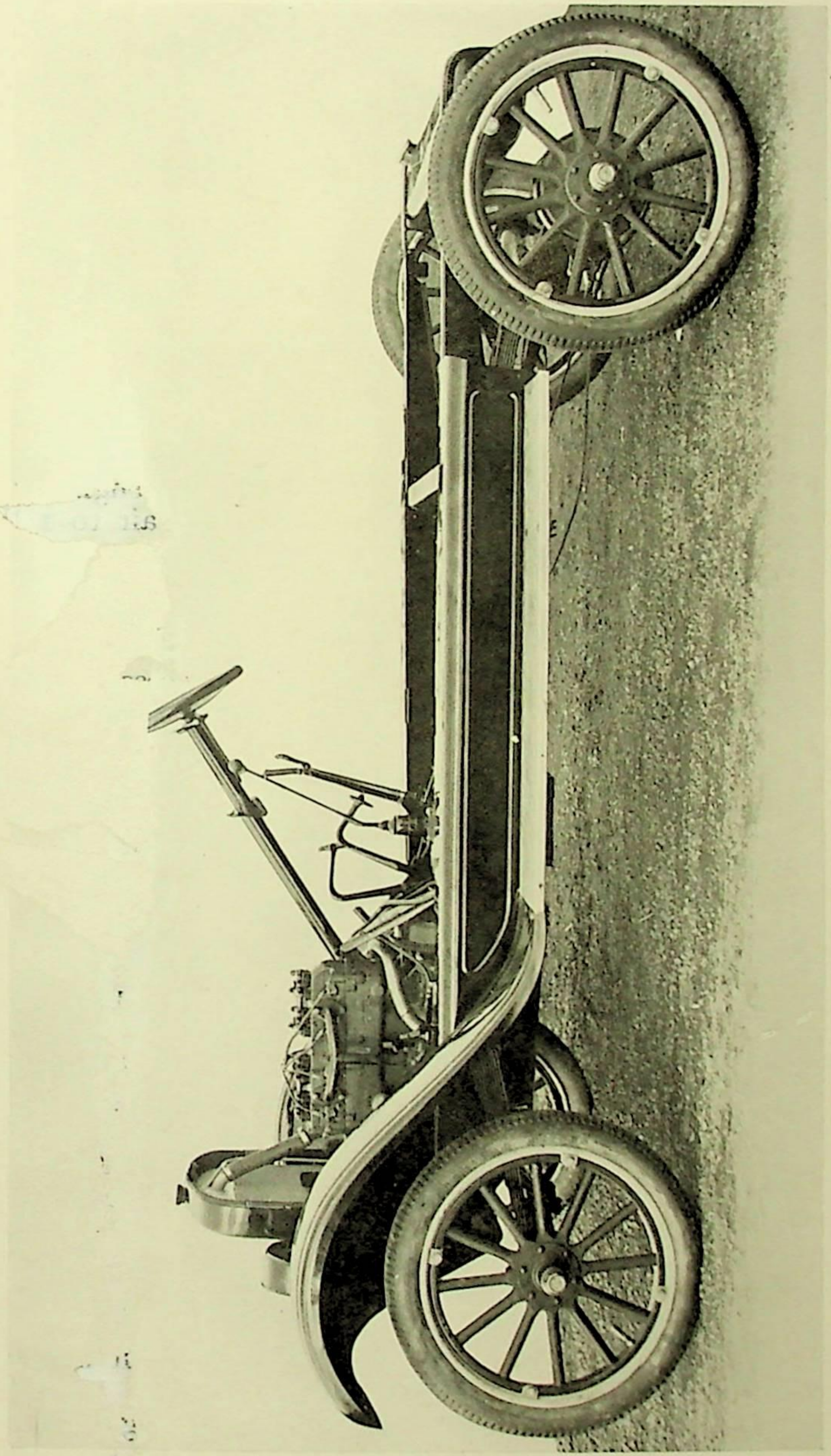
As petrol is both volatile and inflammable it is necessary to take special precautions in its handling and storage, and the official regulations emphasise the fact that a garage or private petrol store must be at least 20 feet from the nearest building whether inhabited or not, unless the petrol is stored underground in a properly constructed tank. This system will be discussed in a later chapter. Not more than 60 gallons must be stored in any one place, at any time, according to the regulations.

BENZOLE

During the past few years this fuel has increased in popularity owing to certain advantages which it possesses, for car-engine use. Benzole is one of the distillation products of the coal tar, obtained during the manufacture of coal gas. The portions which distil at temperatures of about 80° to 100° C. correspond to what is termed "90 per cent." benzole and consist of a mixture of benzene, toluene, and xylene in the proportions 70 to 75 per cent. benzene, 22 to 24 per cent. toluene, and 4 to 6 per cent. of xylene and other hydrocarbons. Its density varies from 0·87 to 0·89, so that it is heavier than petrol. Although, theoretically, pure benzene requires less air than petrol, yet, owing chiefly to its admixture with other diluents, it actually appears to require from 5 to 10 per cent. more air than petrol when it is used in petrol engines.

Owing to its heavier density, its use necessitates weighting or lowering the float in the carburettor, otherwise the petrol level will be too low. Benzole at temperatures of – 4° C. (i.e. 4° below freezing point) begins

PLATE VII



THE 22 H.P. CHEVROLET CHASSIS.

to deposit crystals of pure benzene; at − 12° C. it solidifies, so that this fuel cannot be satisfactorily employed in cold regions; an admixture of 25 per cent. petrol will lower this "unsafe" temperature, however, to − 14° C.

The principal advantage of benzole is that its use in a high-compression petrol engine which is prone to pre-ignition and knocking, in most cases cures this trouble; benzole has therefore a higher detonating pressure and temperature. With benzole, higher mean pressures and heat efficiencies have been proved to result; with a rich mixture an increase in mean pressure of about 5 to 8 per cent. and a reduction in fuel consumption per gallon of from 10 to 20 per cent. have been obtained in tests, made upon unaltered petrol engines.

Benzole will stand a higher compression pressure than petrol, and if an engine is specially designed for benzole it will give appreciably more power than if an ordinary petrol engine be used.

Some doubt has been raised in the past as to the effect of benzole upon the various parts of the car and engine, so that the following notes may be of interest in this connection:

(1) Benzole will dissolve most of the resin varnishes employed on the coachwork of cars, and on motor-cycle tanks, so that care should be taken in its use.

(2) Benzole frequently attacks the varnish used on the cork floats of certain American carburettors; the cork then becomes saturated and sinks, thus allowing the petrol to overflow. Special benzole-proof coatings are now procurable for cork floats.

(3) As regards the effect on the engine, in 1919 the R.A.C. carried out a 10,000 miles road test on a Sunbeam car, running on benzole, after which the engine was dismantled and inspected. Beyond a discoloration of the interior of the carburettor, no corrosion effects were observed in any of the engine parts, and no differences could be detected from the usual petrol results. In fact, it was shown that the amount of carbon deposit in the cylinder heads and on the pistons was distinctly less than would have been anticipated in the case of an engine running on petrol. There was, further, no deleterious effect observed on the lubrication of the engine, and the general performance of the engine during the run was excellent.

PARAFFIN

Paraffin is a rather heavier spirit than petrol, and is obtained from petroleum and shales. It has a density of from ·79 to ·85, so that it is lighter than benzole. Its flash point is higher than that of benzole or petrol, namely, from 30° to 40° F.

Used in the ordinary manner in a petrol engine, paraffin gives rather disappointing results, there being a distinct falling off in power and a general tendency to overheat. It is a cheaper fuel than petrol, and its proper employment in petrol-type engines necessitates the use of a special exhaust-heated vaporising and atomising, or "fog production," device. For medium- and low-compression commercial and motor-boat engines, paraffin can be used quite satisfactorily with suitable precautions. It is usual in most cases to start up the engine on petrol, and then to switch over the fuel supply to paraffin when the engine has warmed up. An example of a paraffin type of carburettor and vaporiser is shown in Fig. 246.

ALCOHOL AND FUEL MIXTURES

Alcohol is a spirit obtainable from a number of organic sources. It is of vegetable origin, and exists in two principal forms, namely, the methyl alcohol, or wood spirit (obtained by distilling wood), and ethyl alcohol, which is obtained by the fermentation process from a variety of substances, such as grain, potatoes, sugar-producing products, cellulose products, etc.

Alcohol is sold in the form of methylated spirit, which consists of about 80 per cent. ethyl alcohol, 10 per cent. wood spirit, 0·5 per cent. of petroleum naphtha, and the rest water. It is necessary, with commercial alcohols, owing to Excise regulations, to add another spirit, termed the "denaturant," in order to render it disagreeable and unfit for human consumption. Alcohol in its purest form has a density of ·795 and a calorific value of 11,600 B.T.U's. per lb., which is just over one-half that of benzole or petrol, so that it would be expected to give much less power than the latter fuels. Actually, however, alcohol can be used with much higher compression pressures (up to 200 lb. per sq. inch), and it gives appreciably higher outputs and heat efficiencies than with the fuels mentioned.

Alcohol has a rather wider range of explosibility with air than petrol. It requires about 9 lb. of air to 1 lb. of fuel for complete combustion, and for this reason it is not necessary to provide so much air through the carburettor. In other words, if a petrol type of carburettor is used, a larger jet (about 50 per cent. greater area) should be used.

Moreover, owing to its high minimum temperature of combustibility, about 20° C., in practice, its use necessitates the heating of the carburettor.

Pure alcohol is not used at present, the preference being for alcohol-benzole and alcohol-petrol mixtures.[1]

A successful mixture which has been used consists of pure alcohol 70 parts and benzole 30 parts; another mixture which has been used in aircraft is composed of alcohol 38 parts, petrol 30 parts, benzole 20 parts, with the addition of about 10 parts of toluol and ether.

In some tests which were made upon a Sunbeam engine, it was found that with a medium compression, at 2,000 r.p.m., a mixture of 50 parts of methylated spirit and 50 parts of benzole gave more power than could be obtained with petrol at any engine speed. The only adjustments made were to change the jet of the Claudel-Hobson carburettor and to regulate the air supply.

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[1] An alcohol-base fuel known as *Discol* is on the market, at present.

The London General Omnibus Co. have carried out a considerable number of bench and road tests with benzole-alcohol mixtures in order to ascertain the most suitable mixture for their " B " type bus engines. The best results were obtained with a 50 per cent. alcohol and 50 per cent. benzole mixture, with a compression pressure of 123 lb. per sq. inch. It was found that the higher the percentage of alcohol the higher was the compression pressure that could be used, and the higher the thermal efficiencies. With a compression of 160 lb. per sq. inch a mixture of 70 per cent. alcohol and 30 per cent. benzole gave the best all-round results. With high compression the ill effects of valve pockets were more noticeable at small throttle openings than with low compression. During the road tests it was found that a new fuel tank became porous and corroded after a month's use. Further experiments showed that copper and iron are very badly attacked by benzole, alcohol, and a 50 per cent. mixture of these. Brass, zinc, aluminium, and tin are only slightly corroded, whilst lead appears to be untouched.

In these tests it was found necessary to heat the carburettor in order to vaporise the mixture. Starting was relatively easy, whilst the acceleration and flexibility were notably good. It was found that " pinking " or detonation were non-existent.

Benzole-petrol mixtures have long since been employed in car engines, the benzole being present to about 50 per cent. of the mixture. The presence of the latter reduces the tendency to " knock," gives more power, and a better fuel economy ; the petrol present prevents the fuel from freezing at temperatures near the freezing point of water.

Another mixture which has been used is naphthalene and benzole, and tests have shown that with 15 to 25 per cent. of naphthalene the fuel consumption is reduced by about 3½ per cent.

It is found that there are some liquids, such as aniline and ethyl iodide, which when mixed with petrol in the proportions of 1 to 2 per cent. will eliminate the detonation tendency and enable rather higher compressions to be used.

A successful fuel of this type is the U.S. Hector fuel, consisting of 30 per cent. benzol and 70 per cent. cyclohexane, and tests show that an engine will operate satisfactorily with a compression of 200 lb. per sq. inch on this fuel ; the freezing point of the mixture was 12° F. below that of water.

CARBURATION

Hitherto we have dealt with the subjects of combustion and of the properties of fuels. a knowledge of which is essential to a proper understanding of the subject of carburation.

We have seen that a petrol engine will continue to run upon a fairly wide range of air-petrol mixtures, the richest (in petrol) mixture containing about 2½ times the petrol of the weakest one ; it has also been shown that for complete combustion of the petrol about 15 times the weight of air to that of the petrol is required.

In order to employ petrol or similar fuels in internal-combustion engines it is necessary to provide an explosive mixture of about these proportions at all speeds and under all conditions, and in a condition of proper admixture or vaporisation. The carburettor of the engine is designed to fulfil this purpose, but, as will be shown later, it is a difficult matter to provide the ideal mixture under all conditions of load, speed, air temperature, atmospheric pressure, and humidity.

The simplest form of carburettor for this purpose consists of a jet of fine aperture placed in a constricted tube and kept supplied with petrol at a constant head or level.

Fig. 213 illustrates diagrammatically such a carburettor ; this is the type from which all modern carburettors are evolved. The principal parts of the carburettor will be clear from the diagram. The action is briefly as follows :

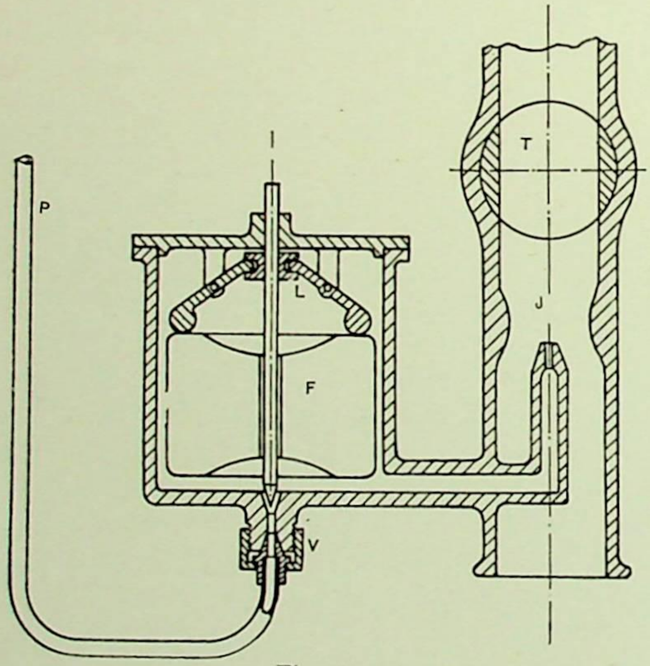


Fig. 213.

Petrol enters the float chamber through the petrol supply pipe P, when the conical needle valve V is opened by the action of the hollow metal float F. When the level of the petrol in the float chamber falls, owing to the petrol flowing through the jet, the float sinks with the petrol and the pivoted levers L drop downwards, forcing the needle valve upward by means of the collar in the needle stem. Immediately the needle valve opens, the petrol flows into the float chamber, raising the level there until the float in rising moves the levers upwards and thus depresses and shuts the needle valve. In this manner a constant level of petrol is maintained in the float chamber and jet J.

When the engine is running, air passes through the main tube shown, and in passing through the constriction known as the *choke tube* around the jet, lowers the air pressure there (due to its increased velocity, which is in accordance with well-known laws of gas flow), and thus causes the petrol in the jet to flow out. In this connection it should be remembered that the pressure on the petrol in the float chamber, namely atmospheric, is greater than the (reduced) pressure in the choke tube. The petrol which flows out of the jet is carried along with the air, in the form of a spray, and forms a mixture of petrol and air in the correct proportions if the carburettor is designed correctly. It is hardly necessary to remark that the air is drawn through the carburettor air port in virtue of the suction created by the descending piston when the inlet valve is opened.

Now, this action is correct only for one speed, and it is here that our simple carburettor shows its demerits. At a higher speed than this, owing to a physical effect, which depends upon the difference between a gaseous flow and a liquid flow, more liquid (i.e. petrol) flows through the choke tube than gas (i.e. air), so that the mixture becomes progressively richer in petrol as the speed of the engine increases, and at a sufficiently high speed the engine will actually choke, unless means are provided to supply the surplus air required.

In the simplest forms of carburettor an additional air supply is arranged, so that when the engine is running above the normal speed for which the carburettor is set to give the correct mixture, the air valve or regulator is automatically or hand operated so as to give the extra air required to keep the mixture constant.

Many motor-cycle carburettors work on the "simple carburettor with extra air device" principle, whilst, on the other hand, certain motor-car engine carburettors often give better results if an extra air device is fitted.

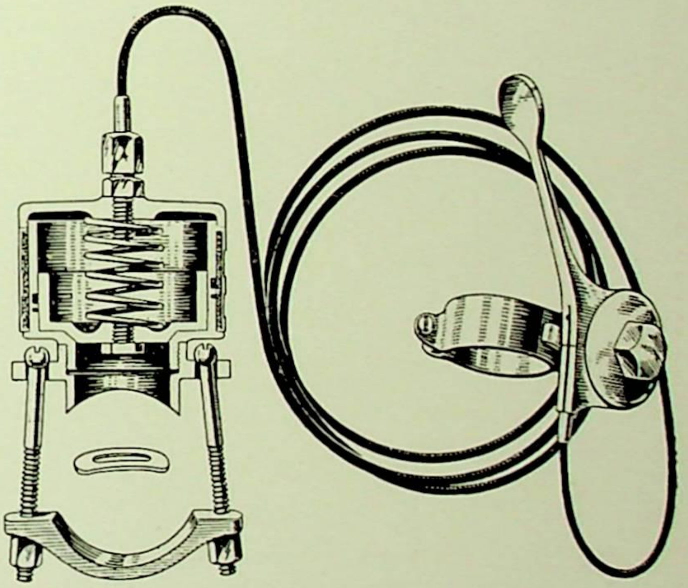


Fig. 214.—The Bowden Extra-air Valve.

An excellent extra-air device of this type is the Bowden pattern, which is controlled from the driver's seat through the agency of a Bowden cable. It consists of one piston sliding in another, so that when the inner piston is pulled by means of the cable it uncovers ports in the outer piston and allows air to pass through and into the engine. A compression spring forces the inner piston back when the cable tension is released.

A sectional view of this device is shown in Fig. 214, which also illustrates the piston valve and spring and the securing clip and screws. In the latest pattern shown, a circular chamber contains a wick which is protected both inside and outside by wire gauze. By saturating this wick with petrol, easy starting can be obtained when the engine is cold. This extra-air and easy starting device is clipped on to the induction pipe, or mixing chamber, above the carburettor.

Another kind of extra-air valve takes the form of a hollow flange which is inserted between the inlet and carburettor flanges. This form is easy to fit and generally necessitates only longer bolts for these flanges. Fig. 215 illustrates the Whalley extra-air valve, which is a good example of this type. It is operated by a Bowden cable from the driver's seat. In the case of some carburettors, instead of working the extra-air control by hand, a light spring is fitted, so that when the engine is at rest, or is running slowly, the valve is kept closed by the spring. When the engine speeds up, the extra suction created in the carburettor and inlet pipe overcomes the pull of the spring and opens the air valve ; the extent of opening increases with the engine speed, and so there is approximately a compensation due to this extra air. In practice, however, it is found that the air-valve spring requires a different adjustment for different conditions, such as temperature and engine wear, and is not satisfactory under these circumstances.

A simple form of extra air device may be made up of a length of copper or rubber tubing from the dashboard to a convenient part of the inlet pipe, and fitted at the former end with an ordinary gas tap of suitable bore. A pipe of $\frac{3}{8}$-inch bore will usually be found to give good results.

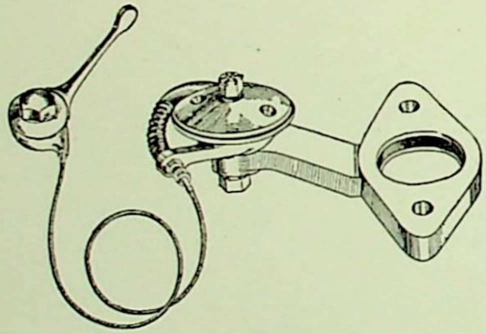


Fig. 215.—The Whalley Extra-air Valve (Inlet Flange Type).

Having considered the simple form of carburettor with its extra-air valve for giving additional air at the higher speeds, let us for a moment consider how such a carburettor will behave under practical working conditions.

Firstly, at very low speeds, as when starting up or running slowly, the air flows through the choke tube too slowly to create the partial vacuum necessary to cause the petrol to flow, and, moreover, in accordance with the physical law previously mentioned, an excess of air flows at lower speeds, with the net result that the mixture is much too weak in petrol to be explosive. Thus the simple carburettor is here seriously at fault.

Again, suppose the engine throttle to be fully opened suddenly when the engine is running slowly, the inertia of the petrol will cause it to lag behind the air, and momentarily the mixture will become weak—in many cases too weak to fire, and the engine will stop.

It will thus be apparent that the simple type of carburettor described is by no means satisfactory under different running conditions, although an appreciable improvement may be made by fitting, not only an extra air-supply valve above the jet, but also another controlling shutter for the main air supply below the jet. In practice this combination would work satisfactorily, but would necessitate much attention and skill on the part of the driver.

IDEAL REQUIREMENTS

From what has been said it will be seen that the requirements of an ideal carburettor are, briefly, as follows :

(1) It must supply correct mixture at all speeds as the throttle is opened.

(2) It must provide correct mixture at large throttle openings and low speeds, as, for example, in the case of an engine climbing a hill on top gear at a fairly low speed (not that this is always advisable, however).

(3) It must give easy starting from the cold.

(4) It should enable the engine to run slowly under light loads, as when the car is standing in traffic, etc., in order not to waste fuel.

(5) It must function correctly under different weather conditions, such as atmospheric temperature changes in summer and winter, and low and high barometric pressures, and humidities.

Although no ideal carburettor exists, there are several well-known carburettors (which will be described in detail later) which make an excellent compromise and give very satisfactory running results.

PRACTICAL SOLUTIONS OF CARBURATION

Before proceeding to a consideration of special designs, it is proposed to consider the principles upon which the working of the better types of modern carburettors are based, that is to say, the methods which are employed to attain the actual requirements as enunciated above.

There are four important methods which are employed to give the correct mixture proportions over a wide range of speed (and when it is remembered that some modern engines have a speed range of from 400 to 4,000 r.p.m., this is no mean achievement) which may be enumerated, briefly, as follows :

(1) The hand-controlled air supply, previously described.

(2) The "hydraulic," "submerged-jet," or "well" principle.

(3) The variable-jet or choke-tube type, actuated by the pressure drop in the carburettor.

(4) The mechanical type, in which the throttle valve is shaped in such a way that it also varies the choke-tube opening.

(5) The automatic air-valve type, either alone or in combination with a variable jet.

It is proposed to consider the above classes of carburettor not already mentioned, and to describe typical modern examples of carburettors in each class.

THE SUBMERGED-JET TYPE

This principle is utilised in several well-known carburettors for obtaining a constant mixture strength under most working conditions.

The diagrammatic illustration shown in Fig. 216 will serve to explain the principle of the working of this type.

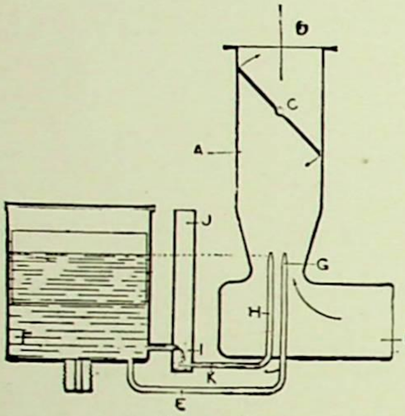


Fig. 216.

It will be observed that there are two jets in the choke tube, one of which is fed directly from the carburettor, is unrestricted, and is termed the *main jet*. The jet, E, behaves in an identical manner to the jet of the simple carburettor. The other jet, H, is connected with a pipe J, open to the atmosphere, and this pipe is fed with petrol through a restricted jet I. Petrol can only flow into the pipe J, and thence to the jet H at a certain rate. If the engine suction in the choke tube is too great, as when running at a high speed, the level in the jet H (termed the "compensating jet") will gradually fall, because the petrol cannot be supplied fast enough owing to the restricted portion at I.

In this manner it will be seen that as the engine speed increases above the normal speed for which the carburettor is adjusted to give the correct mixture, the main jet will, as in the simple type, give too rich a mixture, whilst the compensating jet will be starved, owing to the restricted supply, and the two will thus tend to balance, so as to keep the mixture constant. In practice this method works quite satisfactorily.

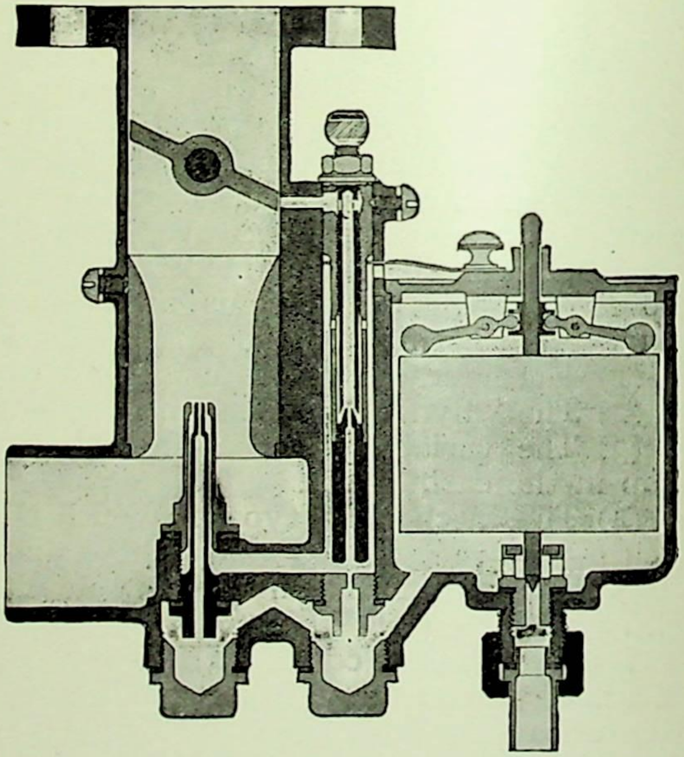


Fig. 217.—The Zenith Carburettor.

The Zenith "ordinary" vertical type of carburettor in Fig. 217 is an example of the application of the submerged-jet principle. The various parts of this carburettor are indicated in the sectional illustration. In this case the two jets shown diagrammatically in Fig. 216 are combined in the form of concentric jets, the inner jet being the main, and the outer annular one the compensating, jet outlets. The constriction in this circuit will be seen immediately above the middle plug. The throttle in this carburettor is of the butterfly-valve type, and opens in a clockwise direction.

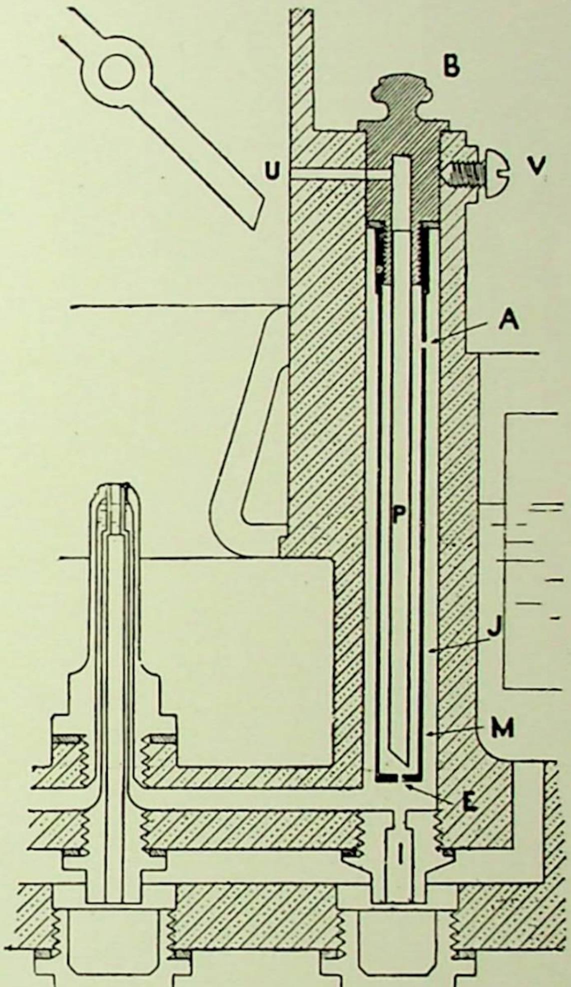


Fig. 218.—Slow-running Device.

The slow-running device on the carburettor consists in principle of a small auxiliary jet in a little choke tube of its own, and communicates with the inlet pipe just above the main choke tube, where one edge of the butterfly valve closes the inlet pipe. It is so arranged that when the throttle is closed the opening to this *pilot jet* is enough for the engine suction to draw a rich mixture from the little jet just described, sufficient in quantity to keep the engine running slowly. This pilot-jet arrangement, which is virtually an auxiliary carburettor, is only operative when the throttle is closed, or nearly so; when the throttle opens, this device is cut out of action, since the suction is insufficient to draw the mixture into the inlet pipe.

Fig. 219 is a part sectional view of the horizontal Zenith type of carburettor for engines having a vertical induction manifold flange. It is exactly the same in principle as the one just described, the only difference being that the mixture, instead of moving vertically past the throttle, turns through a right angle and moves horizontally. The slow-running port to the throttle valve can be seen on the top of the inlet pipe, whilst the air-intake "bell" is just behind it.

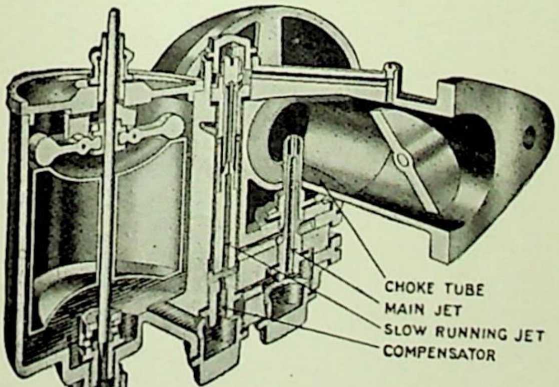


Fig. 219.—The Horizontal Type Zenith Carburettor.

Before describing the improved Zenith carburettor, a short account will here be made of another submerged-jet type, namely, the Solex

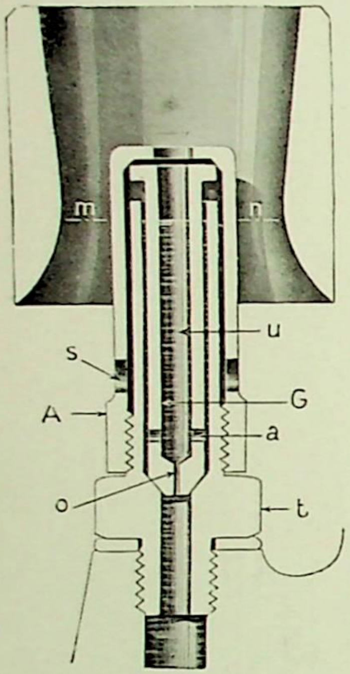


Fig. 220.—Jet of the Solex Carburettor.

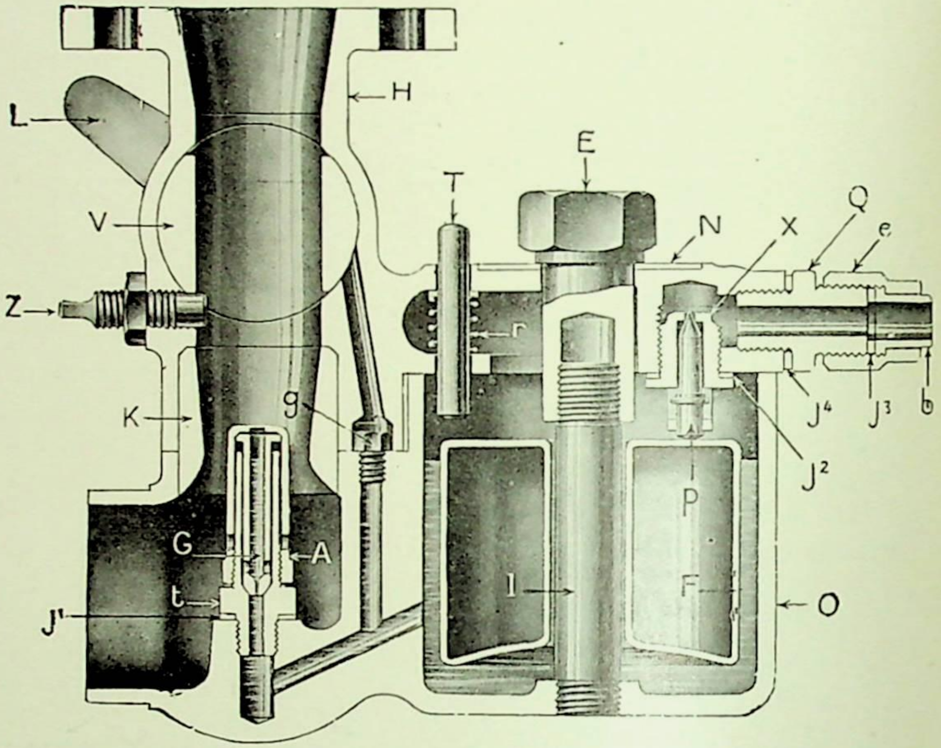


Fig. 221.—The Solex Carburettor, in Section.

carburettor, which is shown in Figs. 220 and 221. The construction of the jet of this type is shown in Fig. 220, from which it will be seen that there is a main jet G, which is fitted with a restricted portion at O, through which all the petrol to the choke passes. This jet is held in place by means of a cap A, which screws on to the lower part of the jet carrier *t*. In this case the submerged jet is 25 mm. below the normal level *mn* of the petrol, and two passages of air ensure that the main jet U will be denuded of surplus petrol as soon as the suction in the choke reaches an

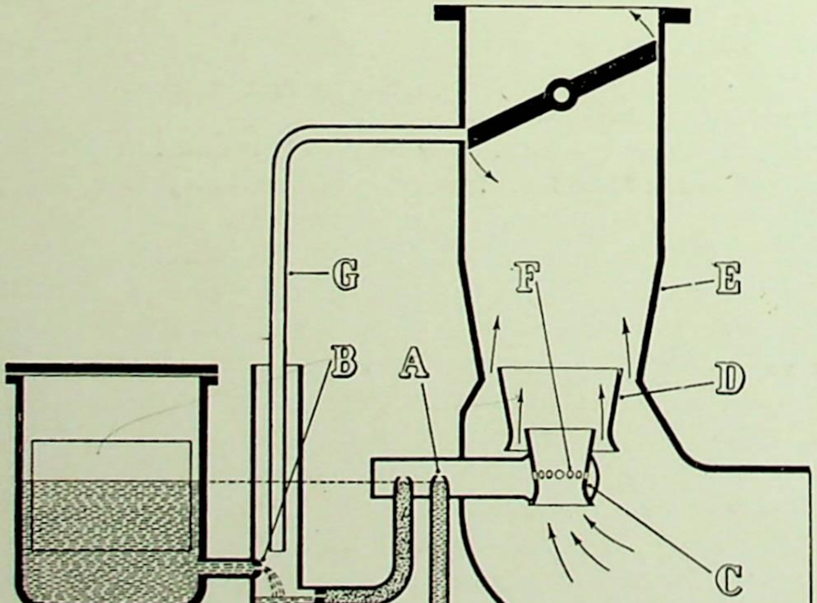


Fig. 222.

equivalent head to 25 mm. of petrol, that is to say, at the lowest firing speed. It will be seen that as the engine speed increases, the level in U will fall progressively so as to compensate for the increasing suction, and the petrol flow will thus be less than in the ordinary unrestricted jet. In this way the tendency to increasing richness with engine-speed increase is counteracted by the falling level in the tube U, due to the restricted portion O. There is also a slow-running pilot jet, Y, similar to the one described previously.

An interesting feature of the Solex carburettor is the ease with which it can be dismantled, and the excellent accessibility when thus dismantled. By removing the large screw E (shown immediately above the float chamber) with an ordinary spanner, the float chamber with its float and two jets can be removed at once. The body of the carburettor remains fixed to the cylinder block, so that it is not necessary to interfere with the petrol piping or unions in order to clean, or to make an adjustment to, the essential parts.

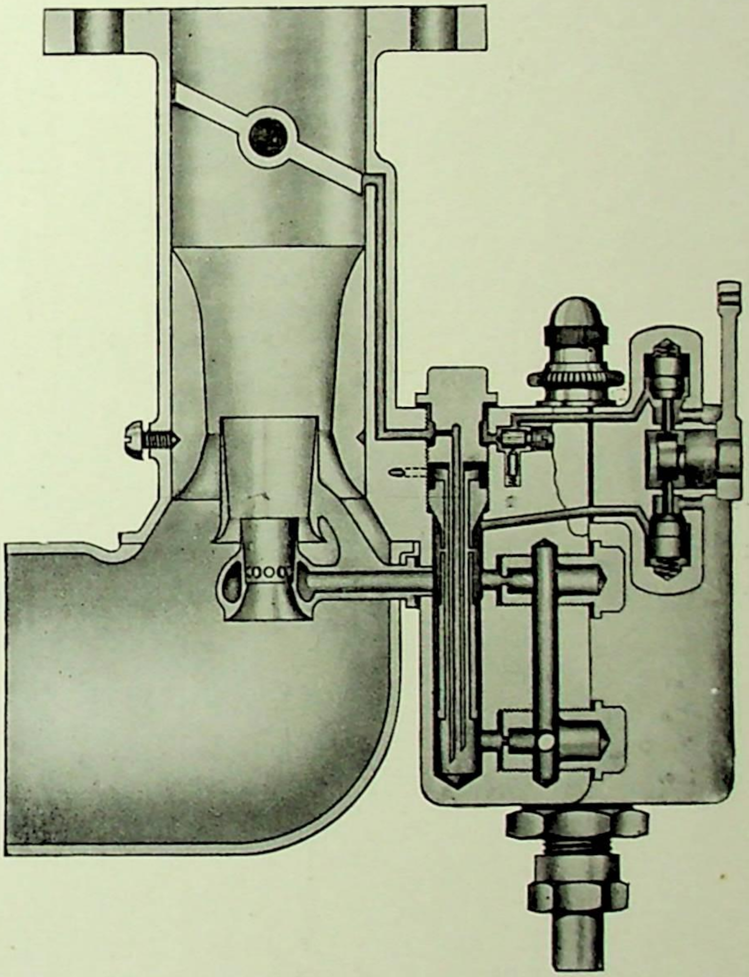


Fig. 223.—The Zenith "Triple Diffuser" Carburettor.

The Zenith "triple diffuser" carburettor, which is shown diagrammatically in Fig. 222 and in section in Fig. 223, employs the submerged-jet principle as before, but in addition an atomising device to break up the petrol from the jets and vaporise it as much as possible. With the modern tendency to heavier-density fuels, in distinction to the lighter petrols of a few years ago, it is important to vaporise or atomise the fuel as much as possible, and to this end the new Zenith carburettor has been designed.

This atomising effect is here obtained by means of high-velocity air, which in its turn is "created" by a system of three concentric choke tubes C, D, and E (Fig. 222). The petrol, after it has been "measured" by the submerged-jet device at B, together with the main jet A, is led to the very small choke tube C through the holes F, and a very great suction is set up, owing to the smallness of the choke tube, resulting in a high-velocity air current which pulverises the petrol as it issues. As there is very little air passing through this diffuser, the resultant mixture is saturated with petrol spray. The air passing through the second choke tube D is not of such a high velocity, but tends to complete the atomising process. The method of construction of this type of carburettor is shown in Fig. 223.

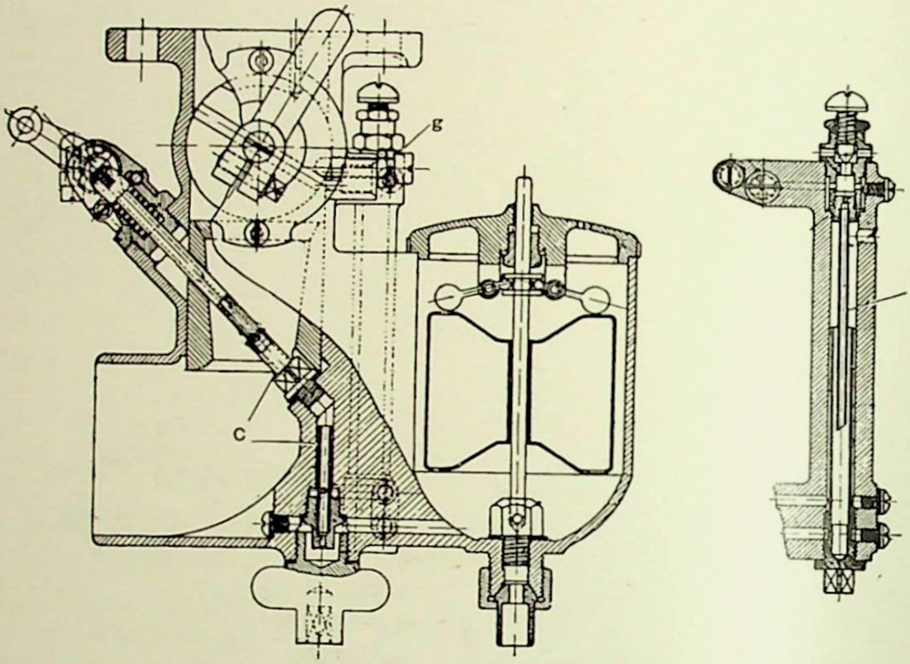


Fig. 224.—The Sthenos Carburettor.

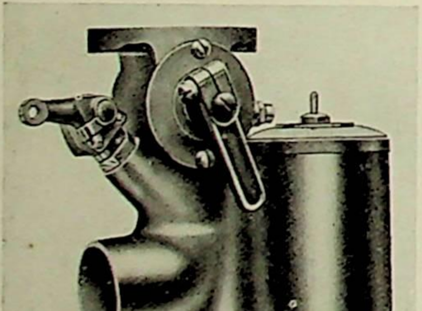


Fig. 225.—The Sthenos Carburettor.

The Sthenos carburettor illustrated in Figs. 224 and 225 is another example of the submerged-jet type, but with certain special features. It has a main jet, C, fitted with a restricted duct and a pilot jet tube, F, forming a well into which a fine-bore tube dips ; this tube also enters the inlet pipe at the level of the throttle, and is provided at its end with a jet of suitable size. In the starting position shown, the throttle is nearly closed, and the petrol level in F is sufficient to immerse the fine-bore tube shown. The strong suction at the open edge of the throttle draws a rich mixture from the tube F for slow-running purposes. As the throttle is opened, the main jet C comes into action, and uses up the petrol in the well of tube F, until at a certain speed the fine-bore tube is uncovered and ceases to supply petrol to the pilot jet, thus cutting it out of action. After this the main jet alone, acting on the *submerged-jet* principle of mixture regulation, is in action.

This carburettor is well designed, as a result of long experience, and all essential parts are readily accessible; for example, the conical jet is held in place by means of the large winged screw shown, whilst the slow-running device at *g* is removed by moving a spring clip.

The Binks carburettor possesses several novel features. It is fitted with a diffuser type of jet, and the mixture regulation over the engine-speed range is attained on the submerged-jet principle. There is also a variable choke-tube device controlled by means of a Bowden cable, which is operated from the driver's seat. The object of this variable choke is to provide an easy means for controlling the strength of the mixture (by varying the velocity of the air past the jet) to suit external conditions of temperature, barometric pressure, and humidity.

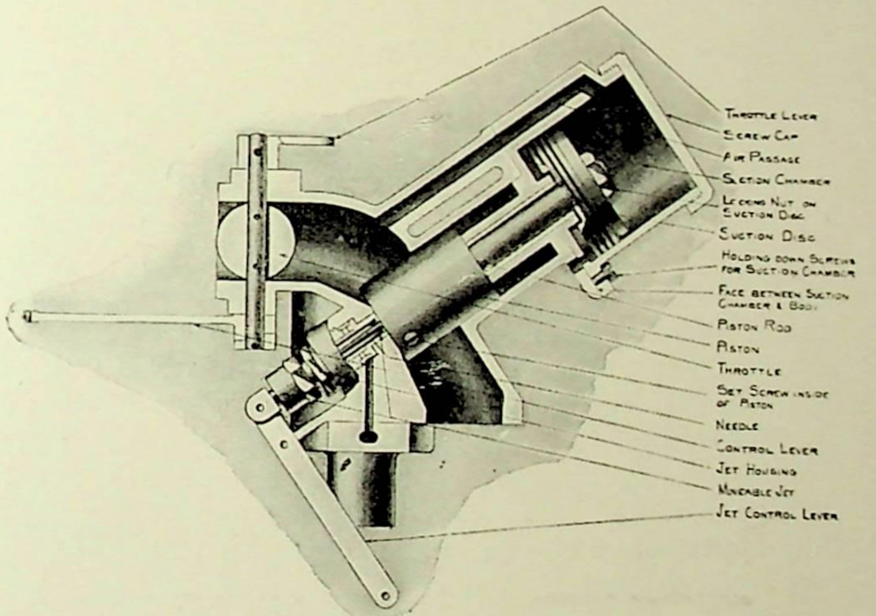


Fig. 226.—The S.U. Carburettor.

The action of the carburettor is as follows: The petrol arrives at the main jet through a passage, and the quantity of petrol delivered by this jet is admitted into the choke tube, a proportion through the retarding sleeve or scroll, and a portion through the other jet. As the suction increases around the diffuser the proportionate flow through the main scroll diminishes and increases through the other jet. This gives a compensating action, for the main jet regulates the mixture and power at slow speeds, whilst the other controls it at high speeds. There is an auxiliary pilot jet provided for starting and slow running, somewhat on the Sthenos lines. The float-needle gear is simple in construction, being of the inverted-cone type, the float being held in position by means of a simple hooped clip.

The S.U. carburettor is an example of the type in which both the size of the jet and the choke tube are controlled by the suction of the engine. The principle of this carburettor is shown in Fig. 226, from which it will be seen that the petrol orifice is fitted with a needle valve, attached to a piston, the lower end of which forms also the choke tube. The piston is actuated by the suction of the engine, and as the engine speed increases it is drawn upwards, gradually withdrawing the tapered needle from the petrol and increasing the area of the choke tube in so doing. At low engine speeds the piston nearly closes the choke space, and thus a high degree of suction comes on the jet, giving a rich mixture for slow running. The relation of the annular jet area and the choke opening due to the piston are so chosen as to give a uniform mixture at all speeds. The latest types of S.U. carburettor are fitted with a controllable petrol jet, the principle of which will be apparent from Fig. 227. It consists of a sliding sleeve; operated by means of the lever L shown, which is worked by a rod, lever, or Bowden cable from a controlling button on the dashboard.

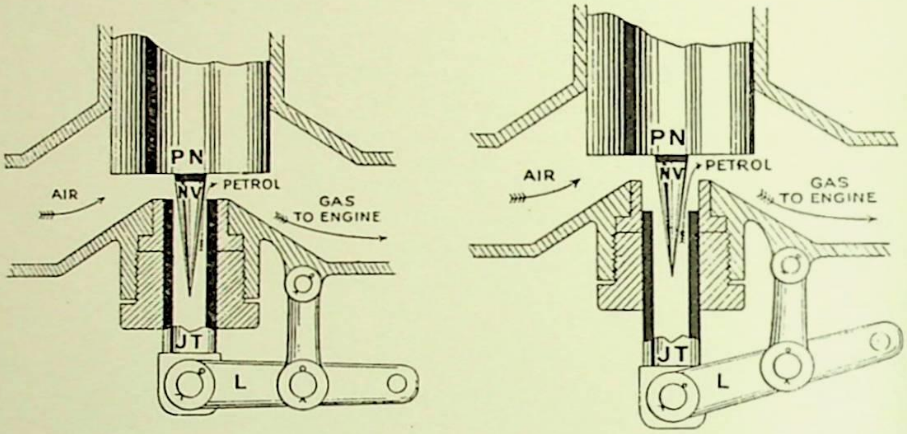


Diagram A. Jet up.
Weak Mixture.

Fig. 227.

Diagram B. Jet down.
Strong Mixture.

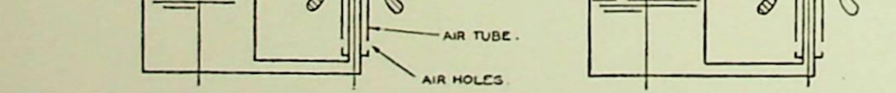


Fig. 228.—Illustrating the Principle of the Claudel Carburettor.

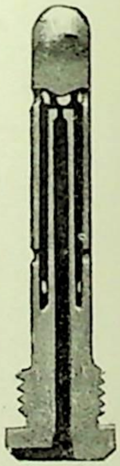


Fig. 229. — Showing Jet of Claudel "A" Type Carburettor.

The Claudel-Hobson carburettor, which is very widely used on automobile engines, is made in two types, the "A" and the "Z" series, of which the former is fitted with a single jet surrounded by a concentric air sleeve, and is termed the *air injector* type, whilst the latter is fitted with a more elaborate diffuser jet.

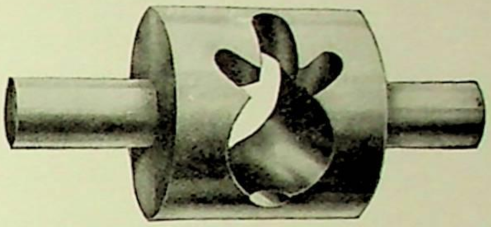


Fig. 230.—Showing Throttle of Claudel Carburettor.

The principle of the "A" type of carburettor will be clear from Fig. 228, which shows the slow-running and full throttle opening positions. A single jet is employed, in which the petrol stands at the same level as in the float chamber, and the top of the jet protrudes into the throttle barrel. The lower edge of the throttle is cut away, roughly resembling an elongated clover leaf, and forms the air regulator for the mixing chamber. An outside view of the throttle is shown in Fig. 230. The jet is surrounded by a concentric tube, or air sleeve perforated with air holes, one set level with the top of the jet, and the other in communication with the air intake of the carburettor. When the throttle is shut, the central slot fits around the top of the jet and its air tube, thus acting as an air regulator, creating a suction on the jet at low engine speeds. As the throttle is opened and the engine speed rises, the suction increases and tends to draw out the petrol in excess amount. This tendency is corrected by the automatic action of the air tube round the jet in conjunction with the special air shapes cut in the throttle. Automatic action is effected by the variation in depression at the top and bottom holes of the air tube, and an atomising action is obtained owing to the fact that the air-petrol mixture issues at right angles to the main air stream.

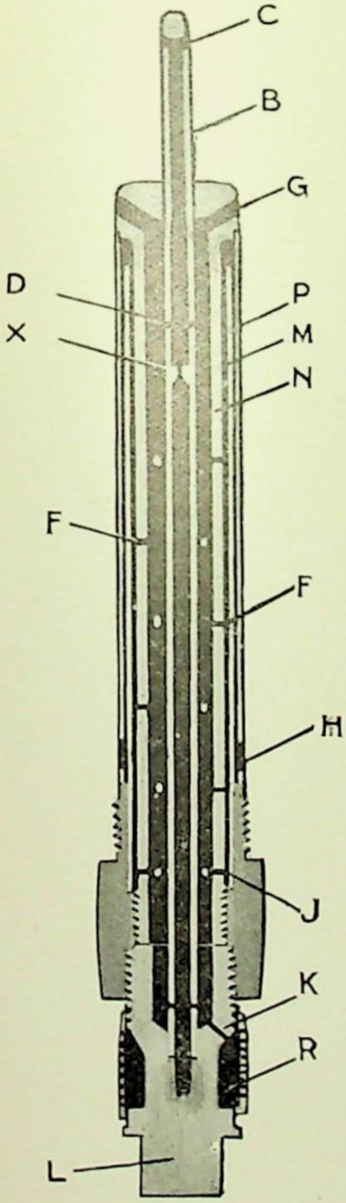


Fig. 231.

In the "Z" or *diffuser* type of carburettor, used on multi-cylinder engines, the main jet is of the submerged type, located at the base of the diffuser assembly, which is shown in Fig. 231. It consists of a calibrated restricted hole, K, in the plug, L All petrol enters the diffuser through the main jet, K, through which it flows into the three concentric tubes B, N, and M. The outer tube, P, is an air-supply tube only. The tube

B is a starting and slow-running tube. The holes JJ near its base admit petrol which has entered the diffuser through the main jet, K. The delivery of petrol to the engine when it is running slowly is by way of the slow-running jet delivery holes CC, and is regulated by the restriction at X, which in reality is a slow-running jet. An air supply is drawn through the holes DD immediately above X.

Fig. 232 is a diagrammatic and Fig. 233 a sectional view of this carburettor. The three tubes N, M, and P provide the correct mixture during normal running. Whilst the engine is running slowly on B, the main jet K admits more petrol than is required for slow running, and in consequence a reserve is formed in the diffuser tube N, and in the guard tube M, which communicate with each other through the holes FF. The level of the petrol in N and M varies continually with the engine speed, and it is the variations in the petrol level in this diffuser which governs the mixture strength, on the submerged-jet principle previously explained. When the engine is running at full throttle opening petrol is drawn out of the tubes N and M as fast as it can enter through the main jet K, so that no reserve can exist, and the petrol level is

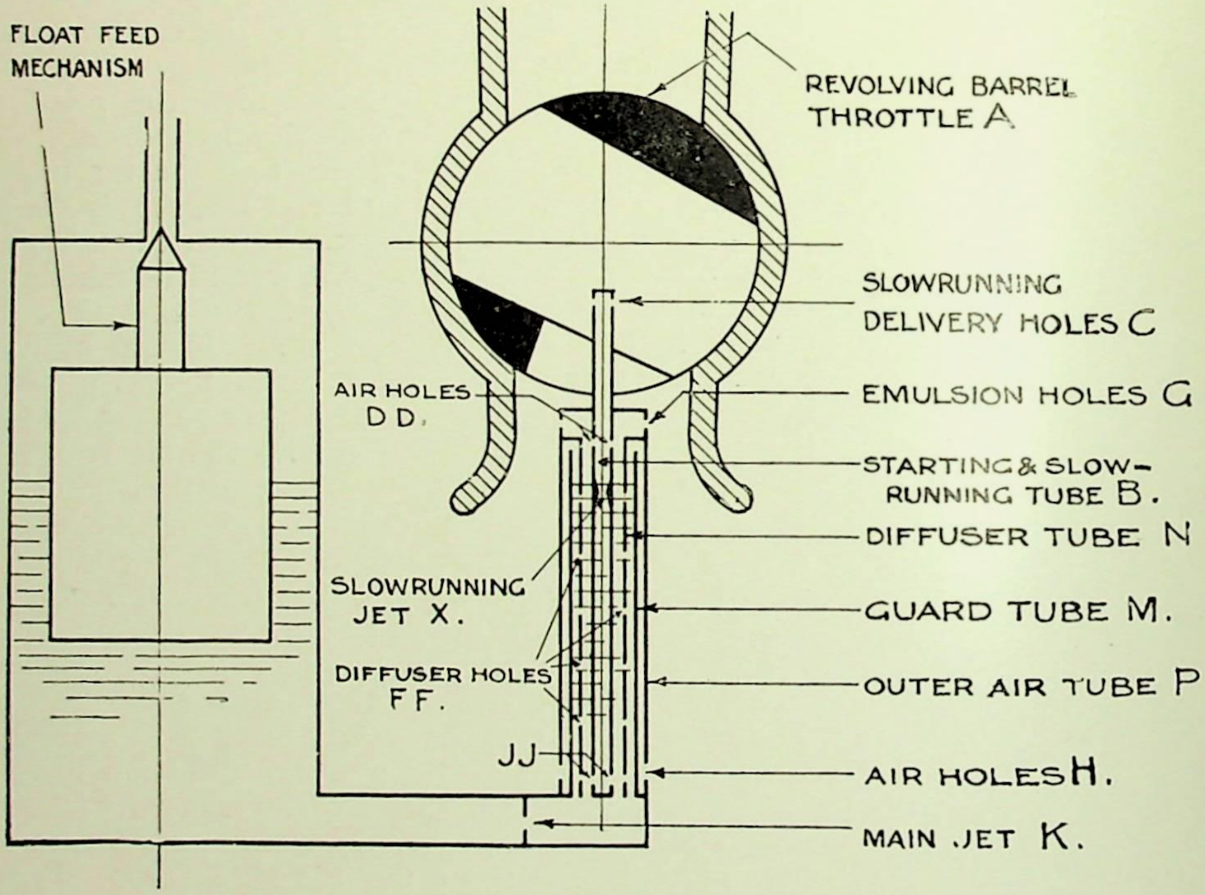


Fig. 232.—Illustrating Principle of Claudel "Z" Type Carburettor.

therefore near the bottom of these tubes. The object of the air holes FF is to reduce the suction on the petrol, by diverting it to the atmosphere as the engine is accelerated. The slow-running jet acts in the same manner as that previously described.

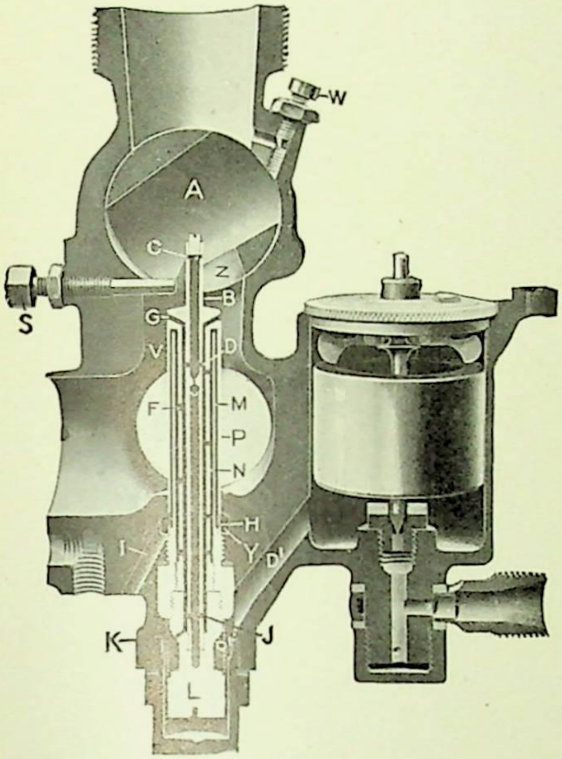


Fig. 233.—The Claudel-Hobson Carburettor (Section).

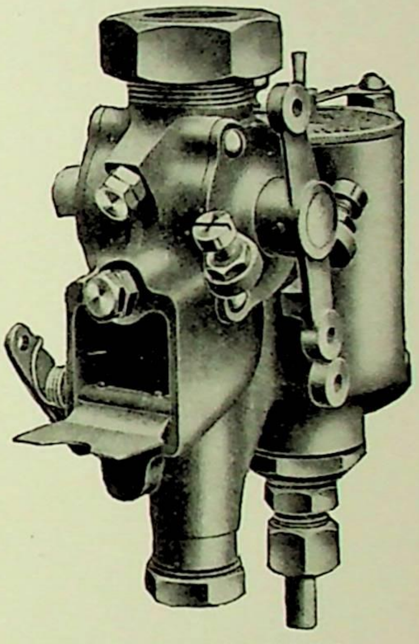


Fig. 234.—The Claudel-Hobson Carburettor (Outside View).

An adjustment pin, S, is provided for regulating the slow-running position of the throttle, and by screwing it in, the mixture is enriched in petrol. The by-pass screw, W, allows of a fine adjustment in the quantity of mixture passing to the inlet pipe when the throttle is closed. In most types an air shutter over the main air-supply entrance port is provided, and is fitted with a control lever.

The White and Poppe carburettor, shown in Fig. 235, is an example of the mechanically controlled type, in which both the air and petrol are regulated by the setting of the throttle. The sectional view in Fig. 235 shows the main jet, A, in the top of which is bored a fine hole, eccentrically, and over which fits a cap, B, having a similar hole in it. The cap, B, forms part of the cylindrical sleeve, C, and when the latter

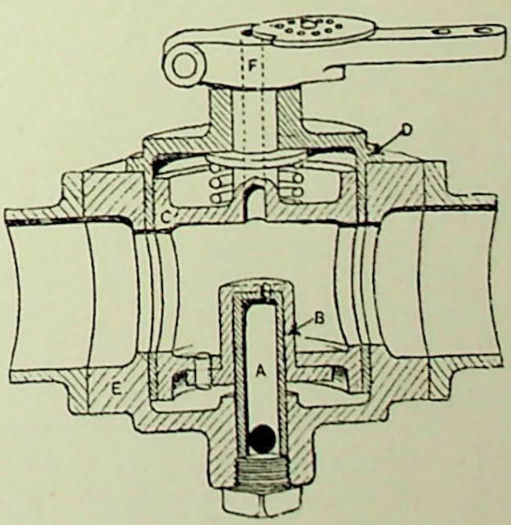


Fig. 235.—The White and Poppe Carburettor (*Autocar*).

is rotated, the orifice between the fixed jet, A, and the cap, B, varies in area. The throttle barrel, C, rotates in another cylinder, D, in which are cut ports corresponding to those in C, and finally the cylinder, D, can rotate in the body, E, of the carburettor.

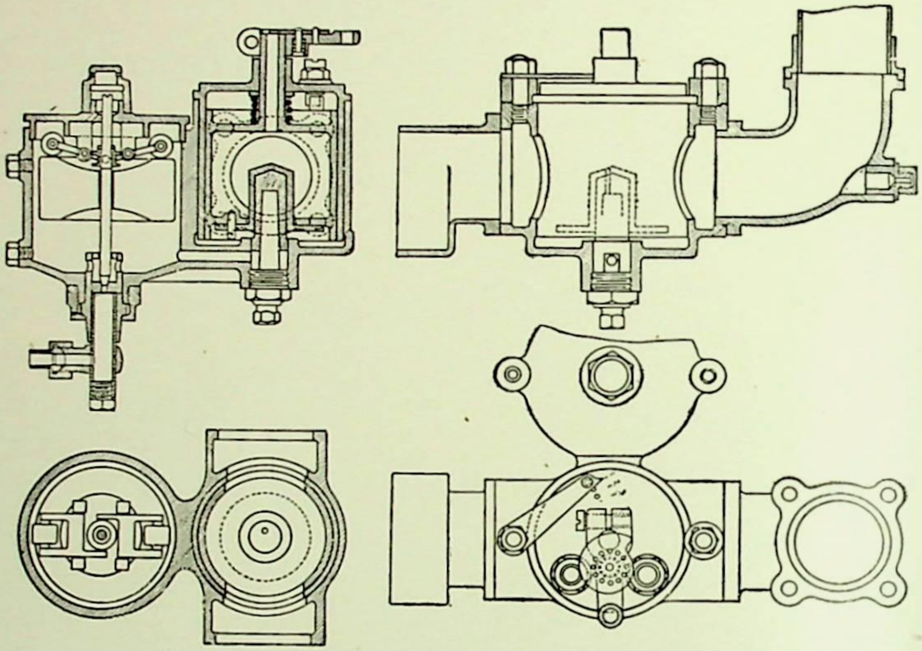


Fig. 236.—Details of White and Poppe Carburettor.

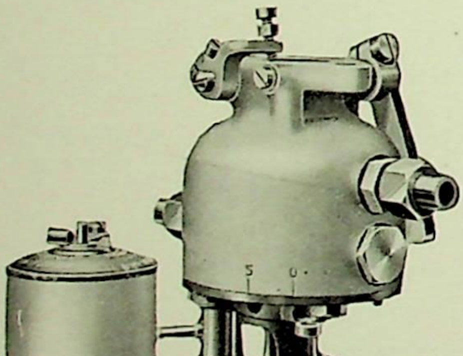


Fig. 237.—The Smith Carburettor.

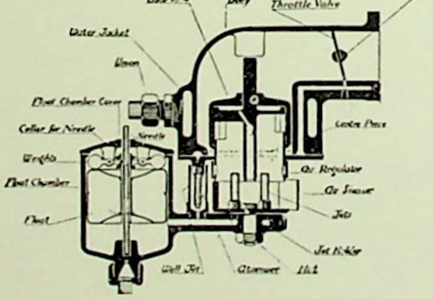


Fig. 237A.—The Smith Five-jet Carburettor.

The rotation of the throttle lever carries C with it and varies the opening of the jet A, and also of the ports between C and D. The amount of air passing across the jet is determined by the throttle opening and by the setting of the adjustable cylinder, D, so that by suitable settings it is possible to vary the air and petrol supplies so as to give a uniform mixture at all speeds. Slow running is arranged for by means of the hole, F, down the centre of the throttle rod, and of which the size of the orifice can be varied by the cam plate shown. This orifice admits air, when the main supply is completely shut off by the throttle, and the extra suction on the jet under these conditions gives a rich mixture for starting and slow running.

Another carburettor which is very widely employed, on account of its satisfactory performance, flexibility, and economy, is the Smith *multi-jet* type, of the mechanical class. In this carburettor, a view of which is shown in Fig. 237, there are five jets in communication with the petrol duct leading from the float chamber, of which four are termed *main jets* and the other a *well jet*. Each of the four main jets has its own choke tube, and the four choke tubes are enclosed in a cylindrical casting, up and down which a piston-type sleeve can slide. Each main-jet choke tube has a port in the cylinder at a different height from the base, so that as the increased suction of the engine acts on the piston sleeve, the latter slides upwards on the cylinder and uncovers in turn each of the four choke-tube ports to the inlet pipe. In reality the ports overlap, so that one is uncovered before the previous one is fully open, so as to give uniformity of mixture control. It will be evident that the four jets and chokes can be accurately calibrated so as to give any range of mixture corrections in the four stages. Thus the mixture can be kept nearly constant by reducing the latter two jet mixtures so as to compensate for the enrichment of the first jets. The fifth, or well jet, is provided for starting up, and for slow running, a hand-controlled (from the dashboard) sleeve being fitted so as to slide up and down the well jet. There is also a rotary air sleeve to control the main air to the four main chokes; this is shown in Fig. 237.

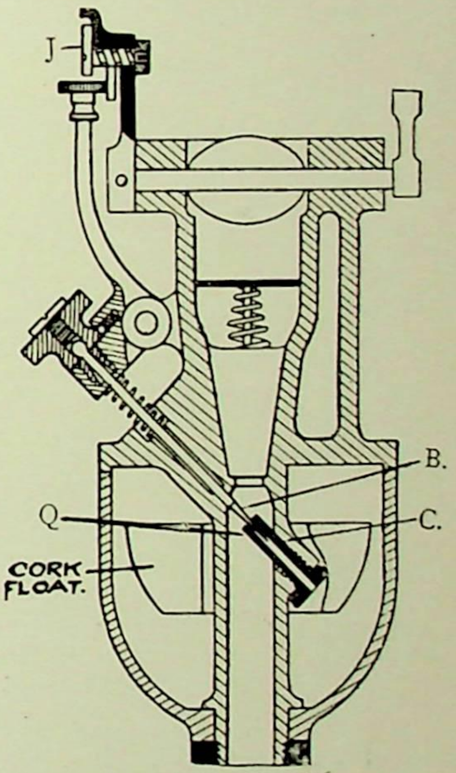
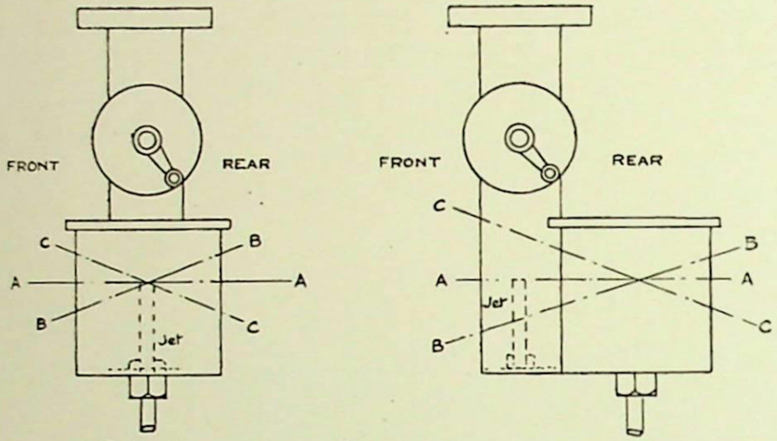


Fig. 238.—The Schebler Carburettor.

An example of a well-known American carburettor is the Schebler, which is of the *variable-jet* type, the variation being controlled by the movement of the throttle lever. The principle of this carburettor is shown in the sectional view of Fig. 238. There is an inclined needle, B, which can move in or out of the petrol supply orifice, C, which is situated in the choke tube, Q. The movement of the needle is controlled by that of a small cam which is mounted on the throttle spindle, J. By shaping this cam in a certain manner for each individual type of engine, it is possible to arrange for any desired variation of the petrol supply as the engine speed increases, so as to keep the mixture constant over the whole range. In another model (namely, the "R" type) of this carburettor an air valve controls the movement of the petrol needle, the action in this case being automatic. Adjustments are included for controlling the tension of the air-valve spring and for the slow-speed needle position. There is also an eccentric which acts on the needle valve, and which can be operated from the dash or steering column, in order to raise the needle off its seating, for giving a rich mixture for starting up. In the "L" type model the throttle-cam-operated needle valve is employed together with a spring-controlled extra-air valve, which opens under the engine suction at the higher speeds.



Correct.

Fig. 239.

A special feature of this carburettor is the annular cork float which controls the petrol level. This float surrounds the cylindrical choke tube, and its use not only permits of a more compact design of carburettor, but also enables the carburettor to work at any inclination without altering the petrol level in the jet.

Fig. 239 illustrates the correct and incorrect methods of fitting the ordinary type of separate "offset" float chamber. In the latter case the level will be too high when descending and too low when ascending hills.

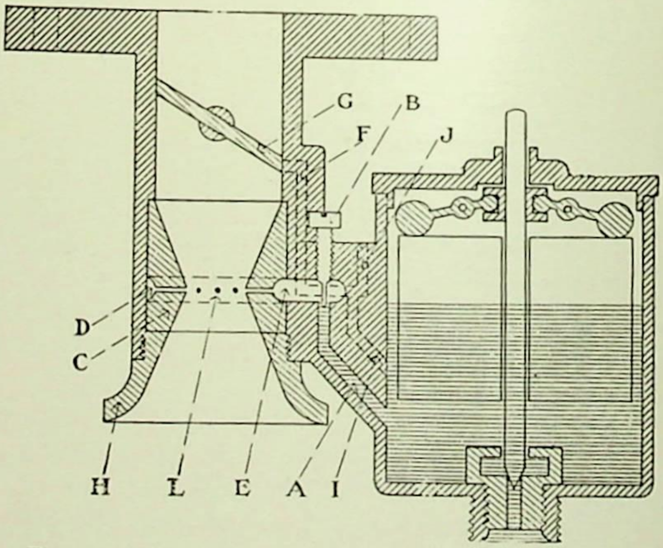


Fig. 240.—The Degory "No-jet" Type Carburettor.

The Degory "No-jet" carburettor is an interesting example of a jet-less carburettor. From the sectional illustration in Fig. 240 it will be seen that, the choke tube being in place, the chambers E and D form but one chamber, the only outlet being the small holes of the choke tube. When the engine is started, the throttle is slightly open and the suction causes a high-velocity air current between G and F, in which an air current starting from J passes over the channel I, forming the mixture there, which then follows the channel F and into the engine; this gives the slow-running condition. As the throttle is opened, the suction ceases in the channel F, but increases in the chambers D and E, and as these have only one outlet (viz. the small holes, F, in the choke tube, H), the fuel passes through these holes in a direction at right angles to the main air current. A regulating screw, B, is provided for the petrol supply, and also one for the zero throttle setting. It will be seen that the submerged jet or well principle of the petrol supply will tend to render the mixture constant with speed increase.

The Brown and Barlow motor-cycle type carburettor is shown in Figs. 241 and 242. There are two controls, which are worked by means of levers on the handle bars through Bowden wires, namely, the throttle and air controls. The jet is situated in a cylindrical chamber in which the throttle and air slides move, each forming one half of a cylinder, so that the two slides completely close the chamber above the jet. The semi-cylindrical throttle slide also carries a tapered needle which works in the petrol jet, so that as the throttle is opened the needle is raised. The needle is specially shaped so that it will give the correct amount of petrol for each throttle position, and the jet is placed on a small choke tube so that a certain amount of air is taken for the purpose of atomising the petrol.

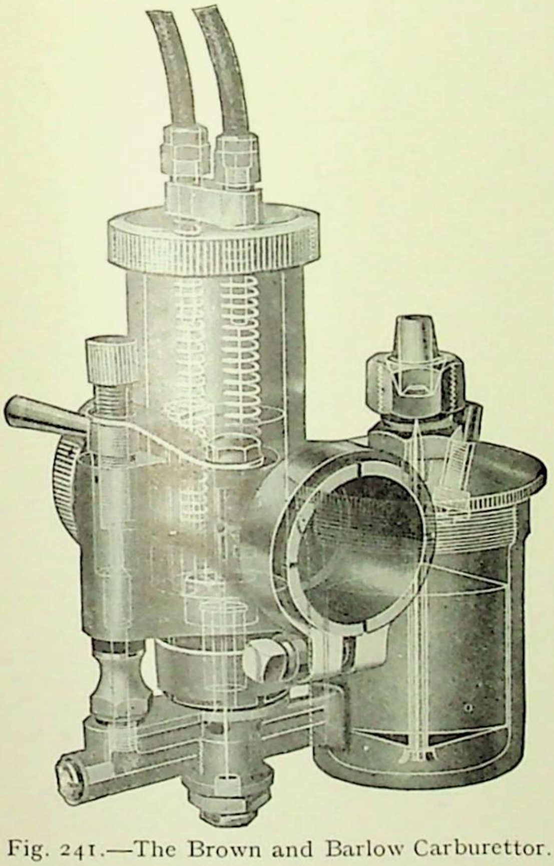


Fig. 241.—The Brown and Barlow Carburettor.

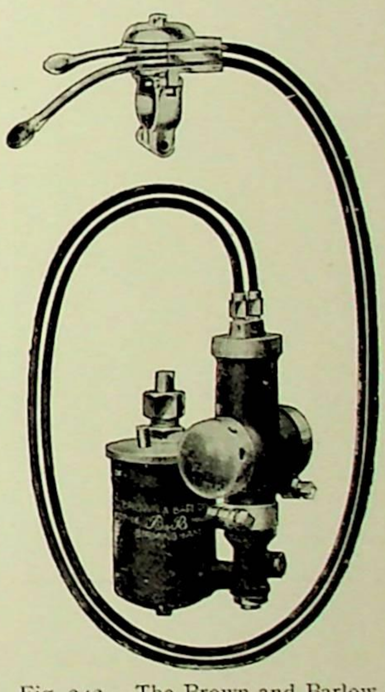


Fig. 242.—The Brown and Barlow Carburettor.

The Daimler carburettor shown in Fig. 243 in the assembled and in Fig. 243A in the dismantled condition is virtually the same in basic principles as the Smith. It consists of several jets in their own choke tubes, each jet being brought into operation by the movement of a sliding sleeve operated by engine suction. By this automatic action the air speed in the vicinity of any one jet is kept fairly constant and an automatic control of the mixture is thus obtained over the whole range of speeds. There is a regulating needle between the float chamber and the jet base by means of which the flow of petrol can be reduced as an aid to fuel economy. For easy starting a priming valve is fitted to the carburettor, which when open allows the engine to draw petrol direct from the float chamber into the inlet pipe just above the throttle valve, so that a rich mixture is provided independently of the other jets. Air to the carburettor intake passes through a pipe having its outer end near the exhaust pipe, so that the air becomes heated, and in addition the carburettor mixing chamber is heated by the hot jacket water.

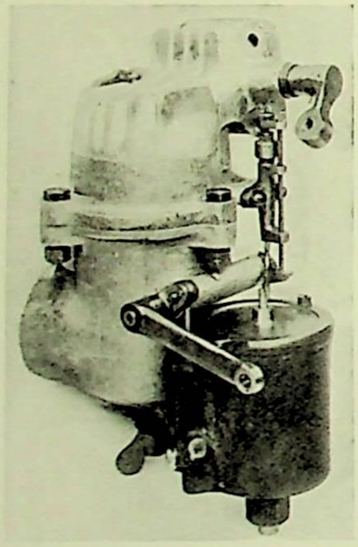


Fig. 243.—The Daimler Carburettor.

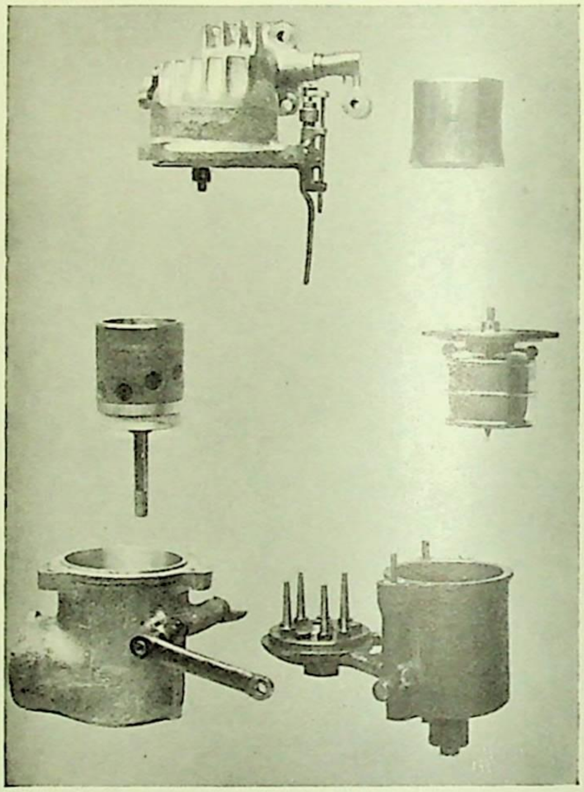


Fig. 243A.

The Wolseley carburettor shown in Fig. 244 is of the floating-weight taper-needle type, with an auxiliary jet for economical running at high speeds. The throttle opening and petrol needle valve shown are controlled by means of a bellows and piston operated by the engine suction, somewhat on the lines of the S.U. carburettor. For easy starting purposes a groove is cut right across the main-jet orifices, so that a high-velocity air current can pass over the jet when the piston is right out. This grooved passage is connected to a point near the butterfly throttle valve. For regulating the mixture according to the throttle opening an auxiliary jet is provided, its effective aperture being controlled by a tapered needle. For about the first half of the throttle opening this jet is closed, all the fuel being taken from the main jet; from this point, however, the cam which is fixed to the throttle spindle allows the needle in the auxiliary jet to be withdrawn, so that as the throttle is opened the aperture is increased. An air shutter is provided for starting purposes.

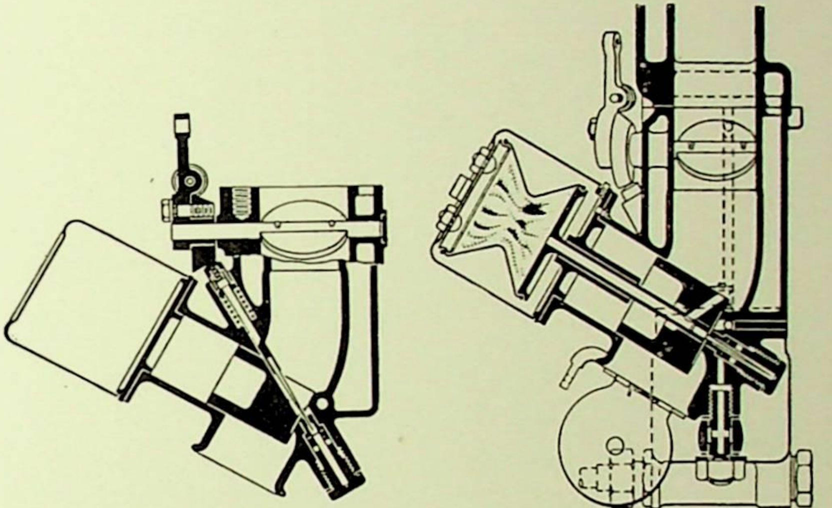


Fig. 244.—The Wolseley Carburettor.

Before concluding this account of some representative carburettors, it may be of interest to mention the Lanchester wick carburettor, which is fitted on a large number of Lanchester cars running to-day; the latest cars are fitted either with S.U. or Smith carburettors. The wick carburettor in question is shown in section in Fig. 245. It consists of a petrol reservoir below, with the wick chamber above, and then a hot-air passage through which hot air is drawn up through the loose wick ends in the upper wick chamber. The mixture leaving the wicks is too strong to be ignited; it is therefore mixed with additional air through the upper air pipe and valve shown in the diagram. The vapour regulator is shown in the intermediate position, the arrows indicating the passage of the air through the various ports. The tank from which the wick sucks is of about one gallon capacity, and is placed just above the supply tank. The wick tank is fed automatically by means of a special petrol pump driven from the engine, so that the petrol is forced into the carburettor directly,

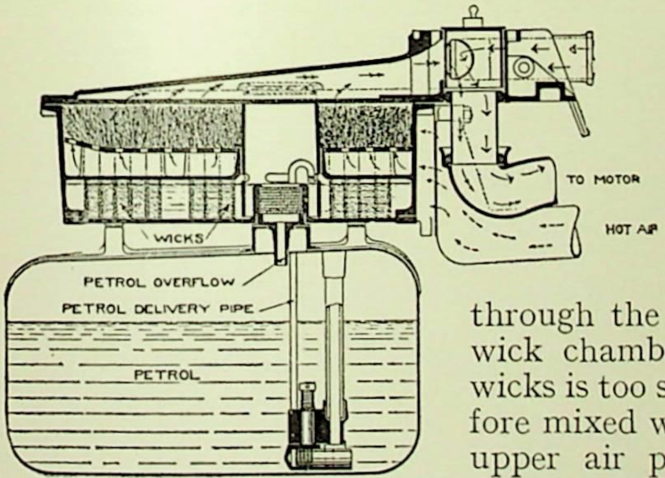


Fig. 245.—Lanchester Wick Carburettor.

and the feed is automatically controlled by the engine speed ; the overflow pipe shown enables all surplus petrol to be returned to the tank below. The petrol pump is shown in the lower tank in the illustration.

One trouble with wick carburettors is that the lighter constituents of the petrol tend to evaporate first, leaving a residue of heavier density behind ; in the jet or "spray" type of carburettor this does not occur.

The Packard carburettor[1] is provided with an adjustable automatic air inlet, and a special and ingenious fuel vaporiser. A sparking plug is arranged in the chamber just above the jet, with gauze inlets to prevent any risk of backfire into the main body of the carburettor. When the throttle is closed the suction of the engine induces mixture through this vaporiser chamber, a small hole being provided just behind the closed throttle position. This mixture is ignited by the spark at the plug, and since they cannot move backwards past the gauze, the hot gases go into the inlet pipe through the hole previously mentioned. The combustion heat tends to vaporise the petrol very thoroughly. The petrol filter, seen immediately below the float needle valve, is of excellent design, and the gauze can readily be removed for cleaning purposes.

PARAFFIN CARBURETTORS

Although paraffin does not give such good results in petrol-type car engines, it is occasionally employed for reasons of economy. The chief difficulty with paraffin is that, although a perfect spray or atomised mixture can be produced, this will not "fire" satisfactorily unless it is pre-heated. Therefore, it is necessary to start the engine on petrol, if an exhaust-heated vaporiser is used, or to use a blow-lamp. The exhaust-heated "mixing" chamber must also be dependent on the throttle position, so as to regulate the heat supply.

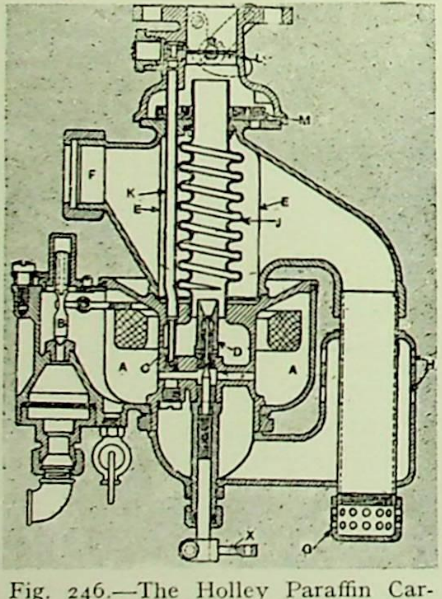


Fig. 246.—The Holley Paraffin Carburettor (*Autocar*).

Fig. 246 illustrates the principle of the Holley paraffin carburettor, which is used on Ford cars and tractors, and also upon other American cars ; the details in the actual carburettor are somewhat different, however. There is a concentric float chamber, A, with a needle valve at B. The central portion of the float chamber carries two jets, one C, for slow running, and the other, D, for normal use. The effective area of the latter is regulated by the needle valve shown, which in turn is controlled by the small lever, X. The mixing chamber, E, is exhaust heated, G being the outlet for the portion of the exhaust gases passing through

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[1] Illustrated in Vol. III. of this work.

E. The main air supply is at H, the main air being heated by its passage around the exhaust outlet pipe G. An extra-air valve is provided at M and admits cold air at the higher engine speeds.

The engine is started on petrol, either by means of an auxiliary float chamber or by an auxiliary tank holding about a quart of petrol and fitted a little below the level of a petrol needle valve and fuel change-over jet. When the engine is running on petrol, after starting up, the exhaust heats the mixing chamber, and as the throttle is opened, paraffin is sprayed from the jet D and vaporises by contact with the spiral tube J. The petrol supply is cut off when the paraffin carburettor has become sufficiently heated to vaporise the paraffin. As previously mentioned, the throttle position controls the amount of exhaust gases which pass through E, a butterfly throttle being fitted in the exhaust pipe, and linked up to the main throttle in such a way that it is fully open when the latter is closed, and gradually closes as the main throttle opens.

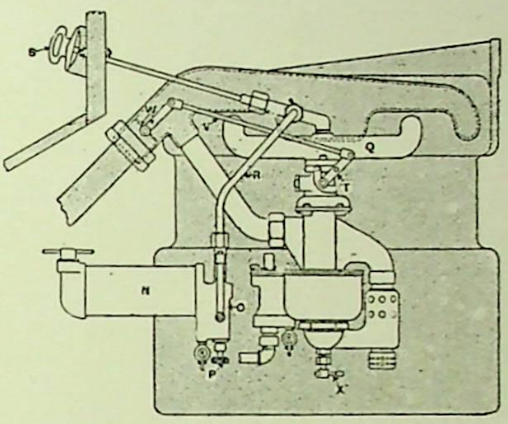


Fig. 247.—The Holley Carburettor fitted.

FITTING OF CARBURETTORS

The size of carburettor required depends upon the dimensions of the engine, and it is therefore usual to specify the size of the carburettor inlet to suit the cylinder bore, although the piston stroke and engine speed should be considered in order to obtain the best results.

It is usual to define the size of the carburettor by the inlet pipe diameter, and this is taken as one-third the cylinder bore for four- and six-cylinder engines, and about two-fifths the bore for a single-cylinder engine.

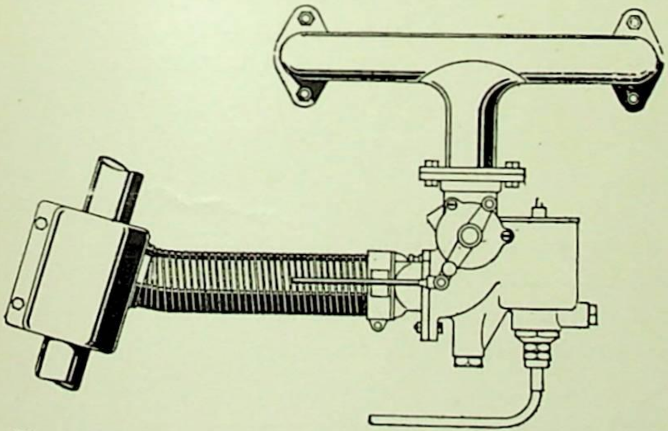


Fig. 248.—Showing Exhaust heating of the Carburettor.

The type of induction pipe employed is very important for vertical-type carburettors. For a four-cylinder engine a good type[1] is that shown in Figs. 248 and 249; the pipe from the carburettor should be straight and the two branches symmetrical. Unsymmetrical pipes for

[1] The best type would have four separate pipes leading into a common pipe at the carburettor mixing chamber.

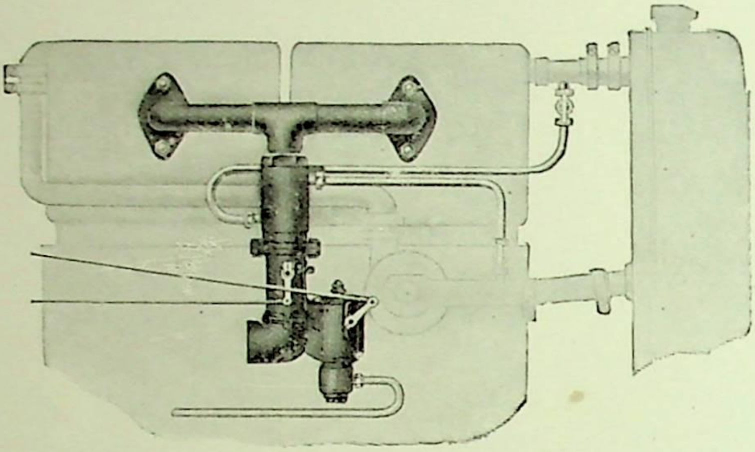


Fig. 249.—Showing Hot-water Jacket in Place.

multi-cylinder engines should be avoided as far as possible, and also those which will reverse the flow of the mixture at one period.

The fitting of two carburettors for racing and high outputs is sometimes done, and in this case an excellent arrangement is that shown in Fig. 250, which gives a uni-directional flow in all circumstances.

Long inlet pipes are sometimes fitted in order to damp out the mixture surgings (or pressure variations) in the inlet pipe, caused by the rapidly moving mixture (the speed often reaches 200 feet a second, i.e. 135 m.p.h.) being suddenly stopped when the inlet valve closes.

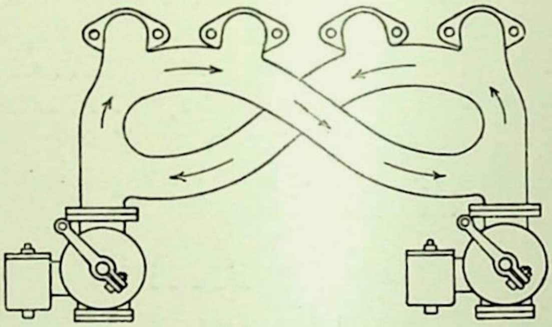


Fig. 250.

The question of the installation of the carburettor involves also that of heating of the inlet manifold, or mixing chamber, or of heating the main air supply for assisting in vaporising the fuel. When the petrol is evaporated from the jet, the lowering of temperature is considerable, and in the winter it is frequently the case for frost or "snow" to be formed on the inlet pipe, and poor atomisation occurs. To obviate this some form of heating is arranged. The two sources of available heat are the radiator or jacket hot water and the exhaust gases.

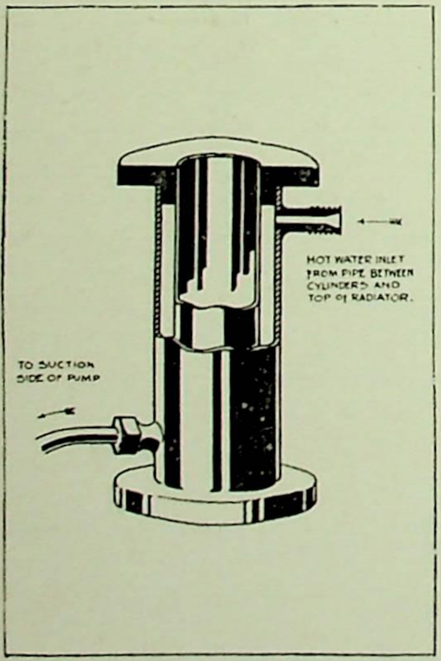

HOT WATER INLET FROM PIPE BETWEEN CYLINDERS AND TOP OF RADIATOR.
TO SUCTION SIDE OF PUMP

Fig. 251.—Hot-water Jacket for Carburettor.

An example of the former method is shown in Fig. 251, and another has already been given in Fig. 249. It will be seen that the inlet pipe to the carburettor hot-jacket chamber (which is simply an annular cavity around the inlet) comes from the hottest part or the main water supply, i.e. the outlet pipe, and the outlet pipe from the carburettor leads to the coldest part, the inlet pipe from the radiator.

The advantage of exhaust gas heating is that as soon as the engine starts, the exhaust pipe becomes hot and the air supply for the carburettor also becomes hot quickly ; with water jacketing, the whole engine must be warmed up before it is effective.

The usual method for exhaust heating is to place a cylinder or muff around the exhaust pipe, as shown in Fig. 248, and to draw the main air supply through this muff. In some cases the inlet manifold of the engine is so designed that it is heated by the exhaust gases in the adjoining or surrounding exhaust outlet passages ; an example of this is given in Fig. 92. In the case of the Morris engines, an air funnel or pipe is cast in the exhaust casing as shown in Fig. 252, and the main air supply is thus heated.

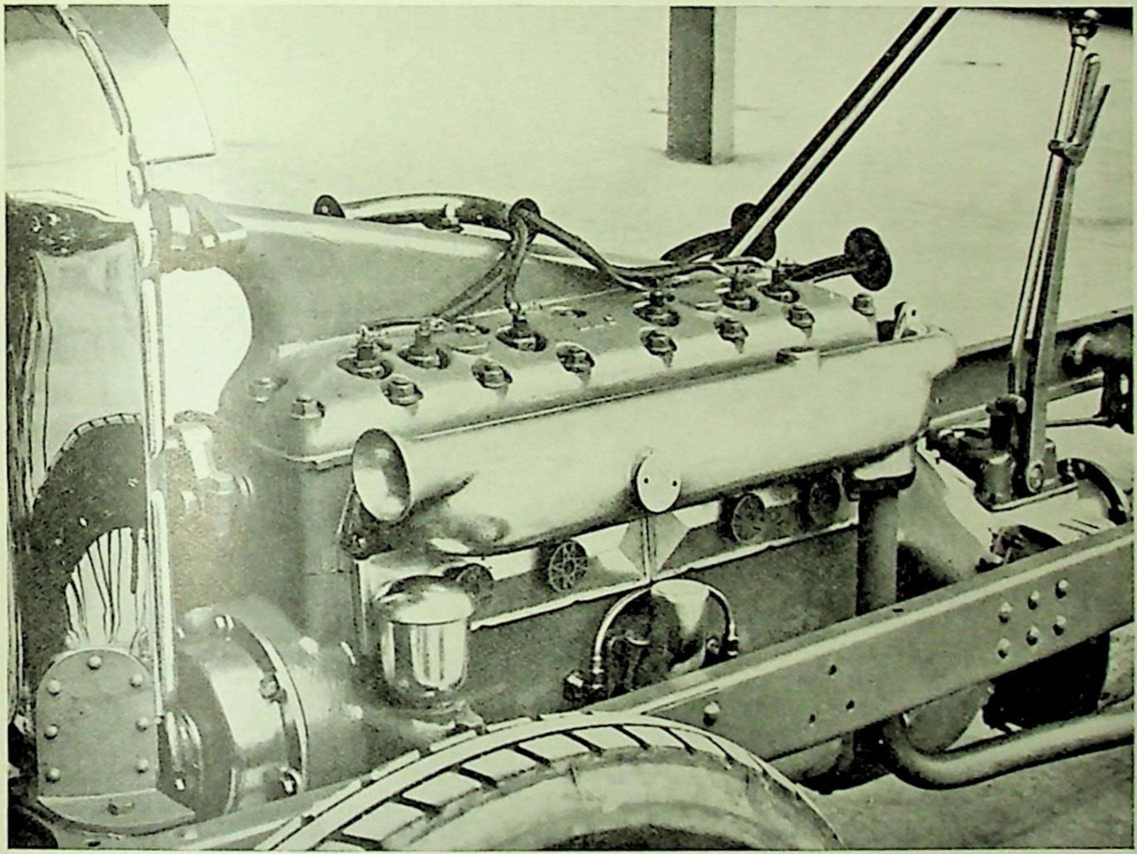


Fig. 252.—Inlet Air Heater on Morris Car Engine.

Many car engines are now arranged with a locally heated inlet manifold, known as a " hot spot," the exhaust gases being the heating means. It is found that better vaporisation of the fuel and quicker acceleration are obtained, although there is often a slight drop in maximum power due to the mixture being less dense on account of the heating. An example of an exhaust-heated ribbed hot spot is shown in Fig. 253; here the exhaust gases pass around a cylindrically ribbed part of the inlet pipe and give the extra heat required for vaporising the fuel completely. When fitting a new carburettor, attention should be given to the question of suitable pipes; all sharp angles and restrictions must be avoided, otherwise "wire-drawing" or throttling of the charge will occur. The rule to observe is to employ uniform section pipes and passages of smooth surface, and with gradual bends where a change of direction is essential.

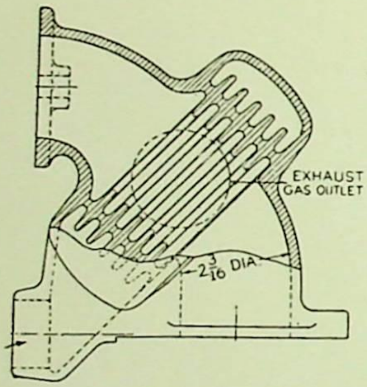


Fig. 253.—The "Hot Spot" Vaporising System.

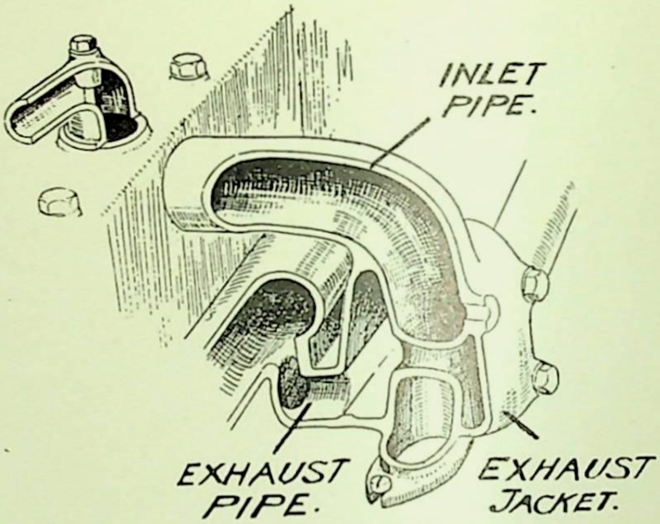


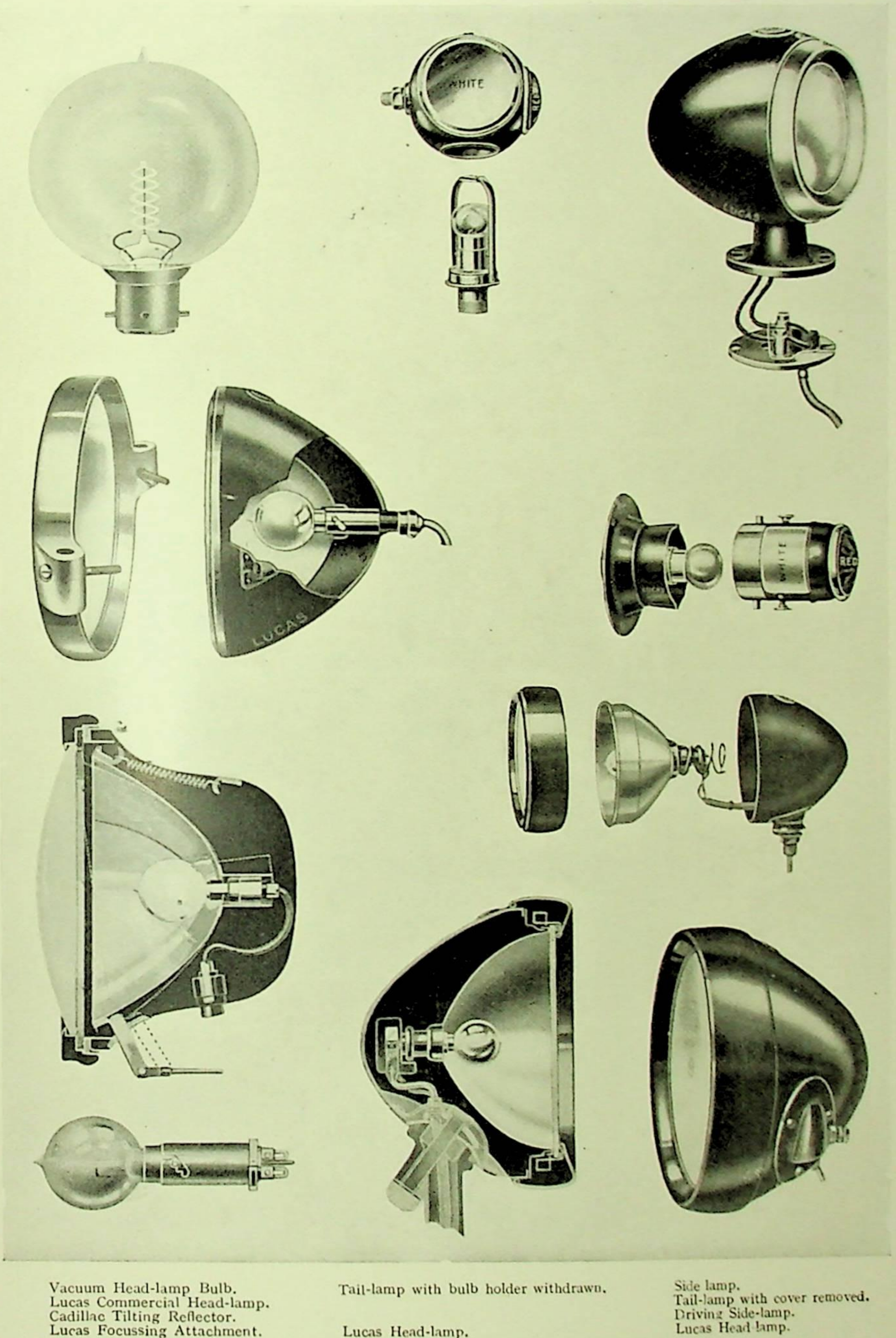
Fig. 254.—Showing Method of Exhaust Heating of the Inlet Pipe.

CARBURATION NOTES

A number of the troubles experienced in connection with the engine running are attributable to incorrect carburettor functioning. The most common fault is the intermittent running of the engine, with failure to respond when the throttle is opened, accompanied by misfiring. This is caused either by water in the petrol, which tends to stop up the jet when the suction increases, or by a choked petrol filter. The filter between the petrol tank and carburettor frequently becomes dirty with the foreign solid matter extracted from the petrol, and should be cleaned periodically.

Back firing, or "popping," in the carburettor, particularly in cold weather, is due to too weak a mixture. If this fault persists in an engine, a larger jet or smaller choke tube should be fitted. In the case of cold-weather "popping," the petrol becomes condensed in the inlet pipe and the amount vaporised is insufficient to give the right mixture.

PLATE VIII



Vacuum Head-lamp Bulb.
Lucas Commercial Head-lamp.
Cadillac Tilting Reflector.
Lucas Focussing Attachment.

Tail-lamp with bulb holder withdrawn.

Lucas Head-lamp.

Side lamp.
Tail-lamp with cover removed.
Driving Side-lamp.
Lucas Head lamp.

More efficient inlet-air or mixture-chamber heating or a temporary shutting down of the main air supply is the cure.

The reason why some engines will not run slowly, even although the compression is good, the valve tappets well adjusted, and the spark satisfactory, is that there are air leaks up the valve stems, and often through the inlet flange joints and extra-air valve seating. The frequent source of this trouble, the valve stems, can be cured by fitting special air-tight sleeves, such as the "Flexikas" type shown in Fig. 255. These sleeves are in the form of volute-springs and are filled with graphite or graphite-grease.

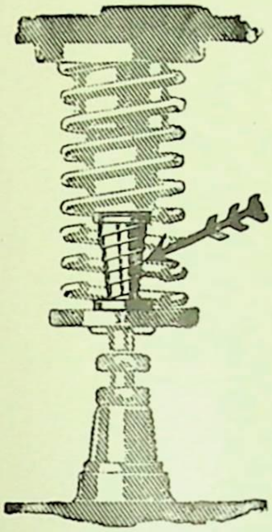


Fig. 225.—The "Flexikas" Valve-spring Sleeve.

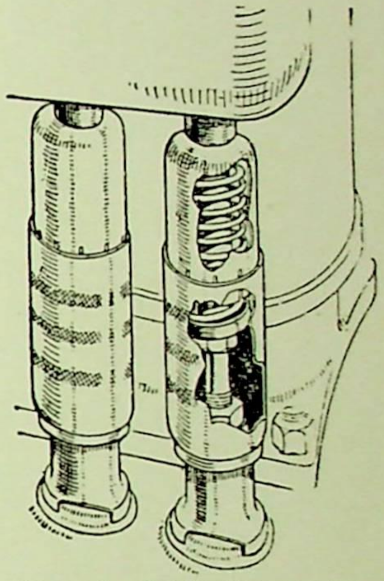


Fig. 256.—Cadillac Valve Casing. with Separate Cylinders.

Excessive fuel consumption is usually a sign of too large a jet or of an insufficient air supply. In summer it is found necessary to provide an additional air supply, or a slightly smaller jet if the engine has been adjusted for winter conditions. If adjusted in summer to run correctly, the mixture will be too weak as a rule in winter.

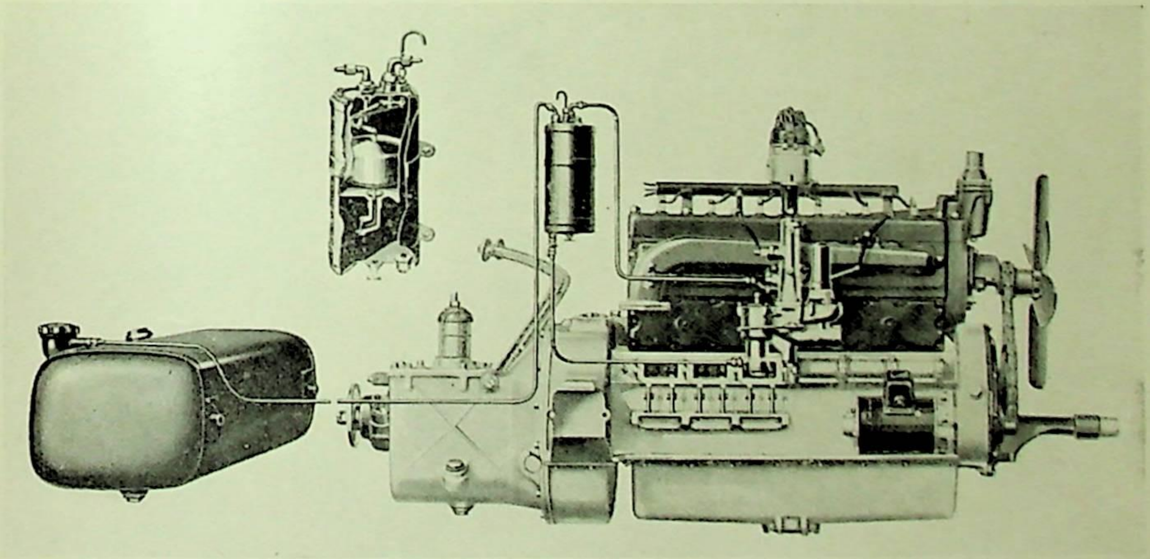


Fig. 257.—The Vacuum Petrol Feed System on the Packard Car.

Difficulty in starting may be due either to a choked pilot jet, to too much air supply, too small a pilot jet, throttle opened too far (so as to cut out the pilot jet), or to the air leakage previously mentioned. Most modern carburettors are provided with main air flaps or shutters and a mixture-regulation control from the dashboard, for slow-running and starting purposes. Failure to attain the usual high speeds is often due to a faulty adjustment of the jet, too small a carburettor for the size of engine, too much heat in the carburettor, or a partly choked petrol supply; water in the carburettor or a choked filter will also give the same results.

FUEL-SUPPLY SYSTEMS

There are three principal methods used for feeding the fuel from the supply tanks to the carburettor, all of which are based upon the general principle of giving a " head " to the fuel at the inlet to the carburettor, in order that it may flow into the carburettor when the needle valve is opened. These methods are as follows:

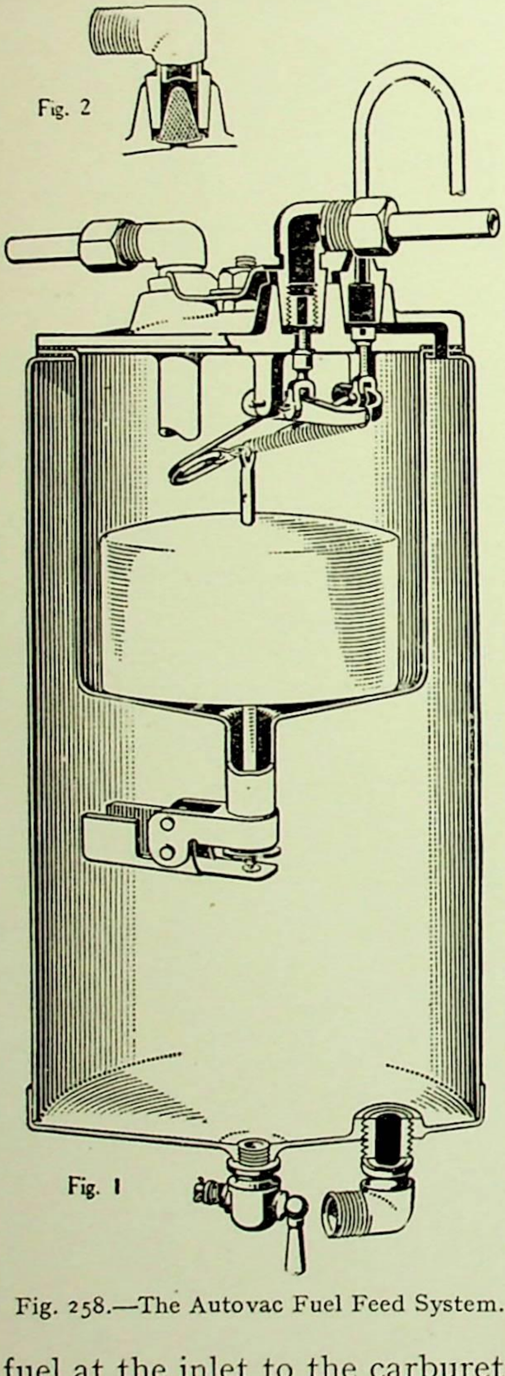


Fig. 258.—The Autovac Fuel Feed System.

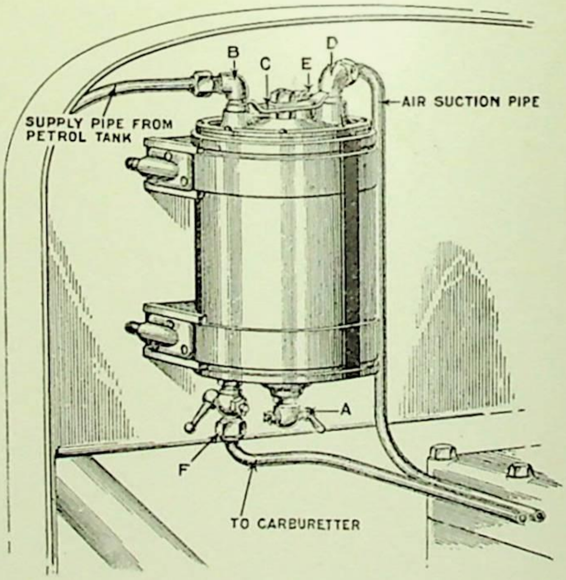


Fig. 259.—The Autovac System on the 12 h.p. Austin Car.

(1) The Gravity Tank, usually situated on the dash. The bottom of the tank must be so placed that on the steepest incline there will be

a minimum "head" of 4 inches, and on the level of at least 20 inches. It is a simple, cheap, and effective method, and is widely employed on the medium-price and cheaper cars.

(2) The Pressure System, in which the main air-tight tank is placed at the rear end of the chassis or under the seats (usually at a lower level than the carburettor), and pressure is applied to the surface of the petrol either by trapping some of the cooled exhaust gas under pressure and leading it to the tank or by means of a small air pump driven off the engine; the latter, method is the superior one. For starting purposes it is usual to fit a hand-operated air pump (with an air-pressure gauge) for giving a sufficient pressure to make the petrol flow into the carburettor.

(3) The Vacuum System, which utilises the reduced pressure existing in the inlet pipe to draw the petrol from the main tank (which is placed at the rear and low down as a rule, for convenience, and is open to the atmosphere) into a smaller auxiliary tank near the engine, whence the petrol flows by gravity to the carburettor. Fig. 257 illustrates the installation of this system in the case of the Packard six-cylinder engine, the main tank being shown on the left, the "vacuum" tank in the centre and above; the inset is a sectional illustration showing the construction of the "vacuum" tank. The upper pipe leading from the "vacuum" tank, on the right-hand side, to the upper part of the carburettor is the inlet "vacuum" pipe supply which transmits the negative pressure to the tank.

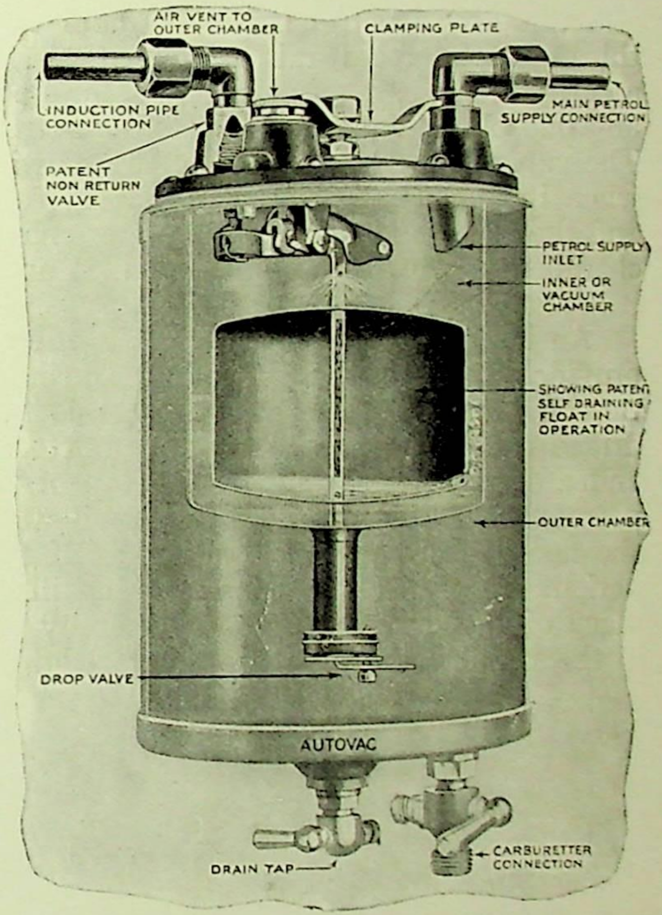


Fig. 260.—The Self-draining Improved Autovac System.

Fig. 258 shows the Autovac "vacuum" tank in section. This system of fuel feed is now widely employed, and has given very satisfactory results; it possesses the advantage that no air-tight joint or cap is required on the main fuel tank, and no special air pump or exhaust pressure-release valve or head pump are required. Since there is always a negative pressure in the inlet pipe, this system can be made quite simple and reliable.

Referring to the illustration shown in Fig. 258, it will be seen that there is an inner air-tight fuel tank which is connected to the inlet pipe at one part of its upper surface, and also to the main fuel tank at another. It is provided with a non-return valve in the inlet side, and also a valve which can open the inner chamber to the atmosphere. The action of this arrangement is as follows: When the engine is working, the induction strokes create a partial vacuum in the upper chamber of the Autovac, and this draws petrol from the main tank. As the petrol flows into this upper chamber it raises a valve float. When this valve float reaches a certain height, it automatically shuts off the vacuum tube and opens an atmospheric valve, which lets the petrol flow into the lower chamber. The valve in the lower chamber drops with the petrol flowing out, and when it reaches a certain point it in turn reopens the vacuum valve and the process of refilling the upper chamber begins again. The lower chamber, as previously mentioned, is open to the atmosphere, so that it acts as an ordinary gravity tank.

An improved type of Autovac is the self-draining float type, which provides an efficient means for ensuring that the float will act, even though punctured. A hollow stem is fitted to the float in which is drilled two small holes, one at the bottom of the stem, *inside* the float, and a corresponding hole at the top of the stem *outside* the float. If petrol should leak into the float, it will automatically be evacuated from the float during the suction period of the apparatus, and during the periods of atmospheric pressure air will flow into the float to take the place of the evacuated petrol, thereby enabling the float to function in the same manner as if it were air-tight.

CHAPTER VII

THE POWER OUTPUT

IT is proposed in the present chapter to consider in both theory and experiment the subject of the horse-power output of the petrol engine.

The horse-power of an engine can be calculated, provided that we know either the mean effective pressure due to the explosion, and the amount of power used up in overcoming internal friction.

The actual horse-power (h.p.) which is generated in the cylinder head is termed the Indicated Horse Power (i.h.p.), whilst the h.p. which is obtained at the crankshaft, flywheel, or clutch is less than the i.h.p. by the amount of the engine friction, etc.; it is termed the Brake Horse Power (b.h.p.), and we have the following relation between the two.

$$\text{i.h.p.} = \text{b.h.p.} + \text{engine losses.}$$

The ratio $\frac{\text{b.h.p.}}{\text{i.h.p.}}$ is termed the "mechanical efficiency," and for a good petrol engine this varies from 80 to 90 per cent.; the higher its value the more useful energy will be obtained from the fuel. It is therefore of great importance to keep down the frictional losses in a petrol engine.

The thermal efficiency of an engine is the ratio of the i.h.p. (expressed in heat units) to the heat energy of the fuel used per minute.

If a weight W lb. of petrol, having a calorific value of C (B.T.U's. per lb.) be used every minute in the case of an engine giving a certain i.h.p., then the following relation holds:

$$\text{Thermal efficiency} = 2{,}540\,\frac{\text{i.h.p.}}{W\,C}$$

The thermal efficiency, as we have already seen, varies from about 20 to 28 per cent., indicating that only this proportion of the fuel energy is usefully obtained as power energy, and since the b.h.p. is less than this i.h.p. in question, the really useful power obtained at the crankshaft will be still less, namely, from 16 to 20 per cent. only of the fuel energy.

If we consider how much of this b.h.p. is lost in transmission, namely, in the gear-box, universal joints, differential and back axle, and in the tyres themselves, it will be found that only some 10 to 15 per cent. of the energy content of the fuel is actually used to propel the car, at the road wheels.

In passing, it should be mentioned that the thermal efficiency, upon which the ultimate road-wheel energy depends, will, within limits, depend upon, and increase with, the compression pressure, the speed of the engine, the weakness of the mixture (up to one consisting of about 17 parts of air to 1 of petrol, by weight), and also with the calorific value of the fuel. It is further found that when an engine is running on light loads, or throttled down, that it is not so efficient thermally as when running on full load.

The horse-power of an engine may be calculated directly as shown on p. 37, but a knowledge of the mean effective pressure is required. The following formulæ, however, are based upon the results of a large number of tests of modern engines, and will give fairly close results at the speed stated, for four-stroke engines.

(1) *Cubical Capacity Formula*:

If V is the cylinder volume (i.e. $\frac{\pi d^2}{4} \,.\, l$, where d and l are the bore and stroke in centimetres, respectively) in cubic centimetres,

$$\text{then b.h.p. at 1,000 r.p.m.} = (7{\cdot}0 \text{ to } 10{\cdot}0)\ V.$$

The higher value corresponds to racing and aircraft engines, and the lower to automobile engines.

For a two-cycle engine, b.h.p. at 1,000 r.p.m. = (10·0 to 14·0) V.

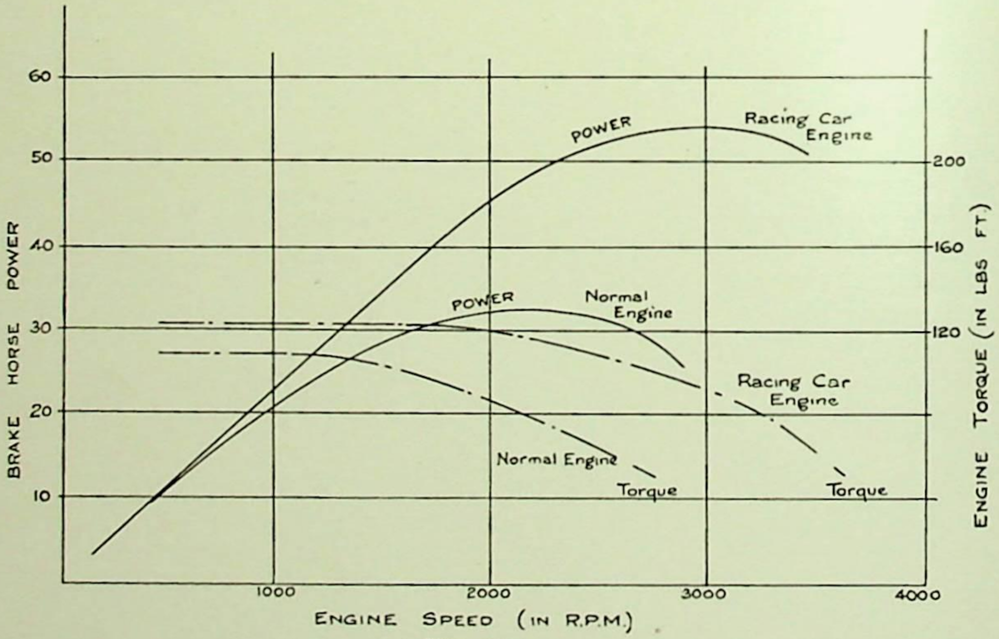


Fig. 261.

The b.h.p. in a well-designed engine increases with the speed up to a maximum of 2,000 to 3,500 r.p.m., according to the type. Thus if the power curve were a straight line up to 3,000 r.p.m., the b.h.p. at this speed would be given by

$$\text{b.h.p.} = (21{\cdot}0 \text{ to } 30{\cdot}0).$$

The results of tests of b.h.p. show that the relation between the b.h.p. and the engine speed invariably follows a curve similar to those given in Fig. 261. The lower curve is for a medium-speed engine as fitted to ordinary cars, and the upper one for a high-efficiency engine.

Fig. 292 shows the power curve, and also the fuel consumption, turning moment, and brake mean pressure curves in the case of the A.E.C. 45 h.p. lorry engine mentioned on p. 100.

The torque is calculated as follows :

$$\text{Torque (in lb. ft.)} = 5{,}250\,\frac{\text{b.h.p.}}{n} \quad \text{where } n = \text{r.p.m.}$$

(2) *The R.A.C. Formula* for h.p., and which is universally used for taxation and (nominal) engine rating purposes, is as follows :

$$\text{b.h.p.} = \frac{d^2\text{N}}{2\cdot 5} \text{ (or } 0\cdot 4\, d^2\, \text{N)},$$

where d = diameter of cylinder in *inches*, and N = number of cylinders.

This formula, which corresponds to a piston speed of 1,000 feet per minute, and to a brake m.e.p. of 67·2 lb. per sq. inch, gives an approximate value of the b.h.p. at about 1,000 r.p.m., but does not take the *stroke* into account, so that long-stroke engines can cheat with this rating. There is a variety of formulæ which have been proposed for the b.h.p., but it will serve no useful purpose to consider them here.

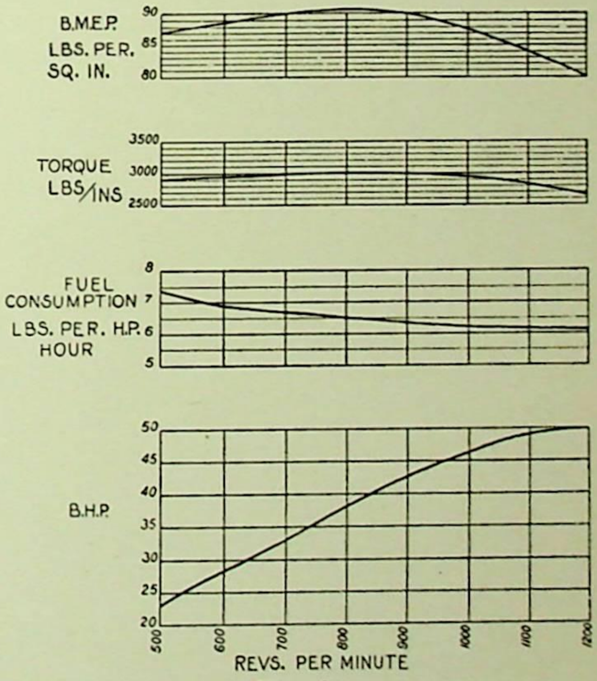


Fig. 262.

The i.h.p. can be found, in practice, by measuring the cylinder pressures at different parts of the compression and explosion strokes, and by averaging up so as to obtain the mean pressure (or m.e.p.). Instruments (known as indicators) have been designed for measuring these cylinder pressures, although considerable difficulties exist in this case, due to the very high speeds of movement of the indicator parts, and to the high temperatures to which the instrument is often exposed.

Amongst these indicators, the Hopkinson and the Watson types are about the best known.[1] The latter type employs a light mirror on fine pivot points. The pressure acts upon a corrugated diaphragm and a

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[1] The Wood electro-pneumatic indicator, now being made by Messrs. Dobbie-Macinnies, is the latest development in indicators.

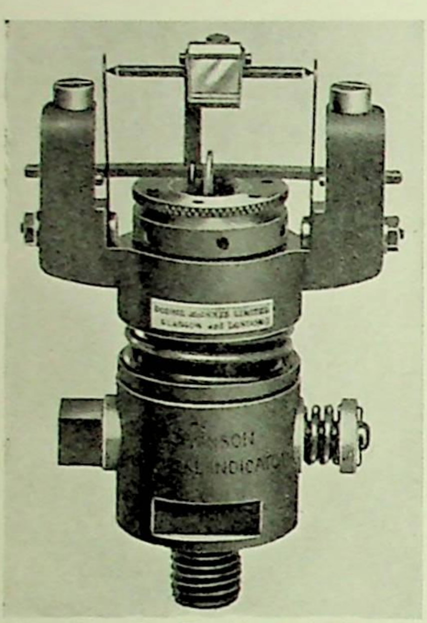


Fig. 263.—The Hopkinson Indicator.

light rod, and tilts the mirror, thus reflecting a beam of light on to a ground-glass screen ; another mirror is also rocked about a transverse axis so as to displace the beam, in a manner similar and proportional to the motion of the piston.

Figs. 263 and 264 illustrate the Hopkinson indicator, which is screwed into the cylinder head. The piston, F, communicates its motion to the mirror, M, which it rocks, through the agency of the beam type of spring, AB, which is fixed at its ends. The whole upper part, X, carrying the mirror, can rotate about the axis of the piston on the ball race, S, being constrained by means of the spiral spring shown. Thus the mirror has two movements or "rockings," one proportional to the cylinder pressure and the other is made proportional to the position of the engine's piston along its stroke. In this way a piston-pressure diagram is obtained, such that it gives, to scale, the pressures occurring at all points of the piston's stroke. The tap T is for the purpose of opening, or shutting off, communication with the cylinder. In using this indicator a beam of light is reflected, from a fixed point source, off the mirror on to a ground-glass screen or photographic plate on which it traces the indicator diagram. Examples of typical diagrams are shown in Fig. 265, A and B, with the piston stroke base marked upon them ; Fig. 265, B, shows typical two-stroke indicator diagrams for the cylinder and crankcase.

An instrument for measuring explosion and compression pressure is shown in Fig. 266. This instrument screws into the cylinder or valve cap, and contains a small spring-loaded piston upon which the pressure acts against the force of the spring. A small lever on the left-hand side is actuated by the movement of the piston, and

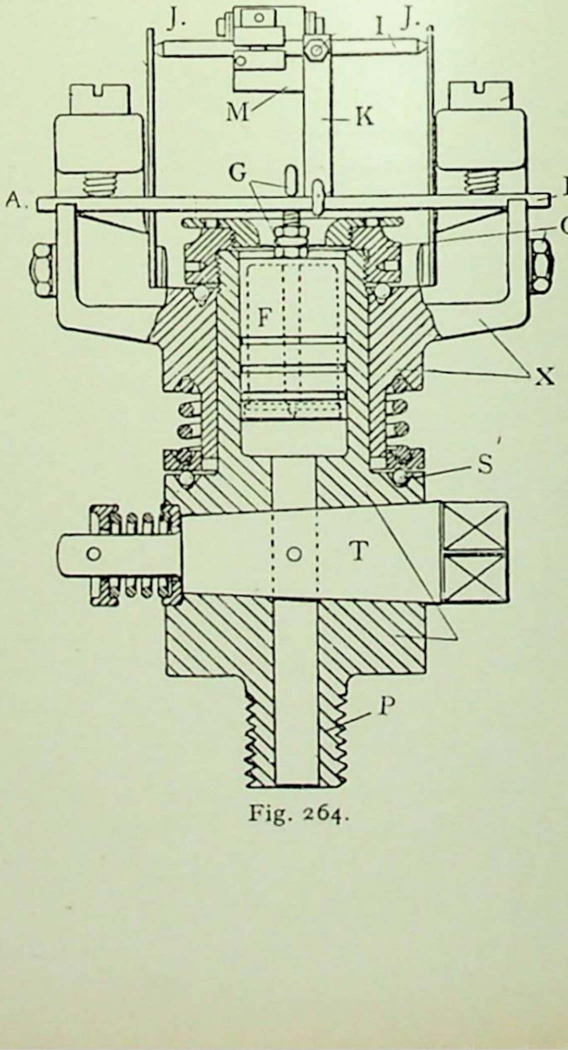
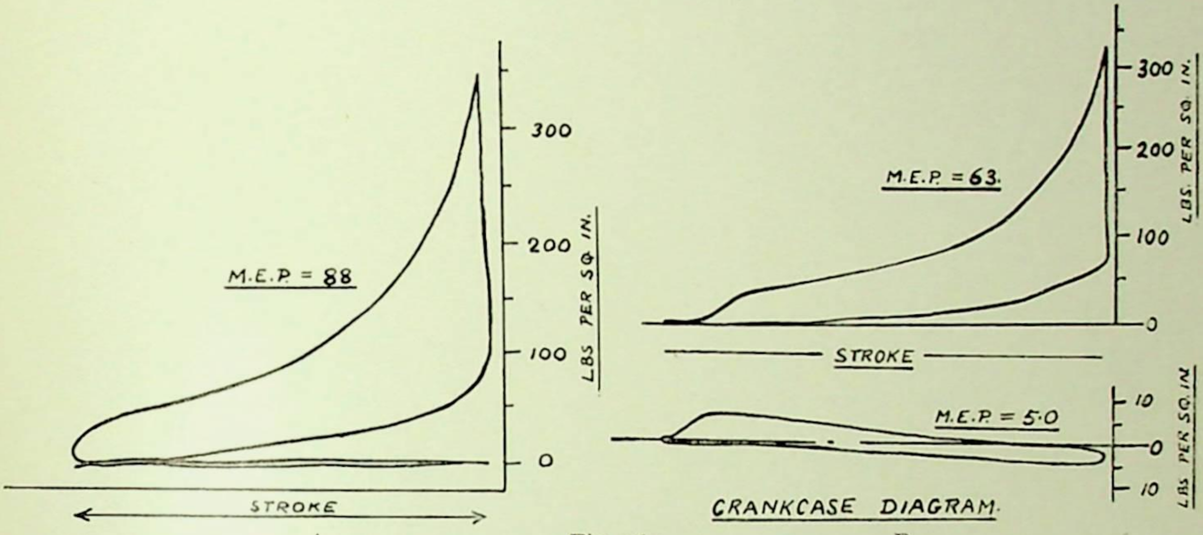


Fig. 264.

works over a scale. The instrument is used by screwing the milled cap D up or down until the spring pressure just balances the explosion pressure, as indicated by the lever on the side ceasing to move. The pressure can then be read off a graduated scale on the side, for the spring load and area of the small piston are known and therefore the pressure can be computed.



A Fig. 265. B

It is not possible to use an ordinary Bourdon pressure gauge for indicating the explosion pressures, although compression pressures, whilst the engine is slowly cranked over, may be measured in this manner. The b.h.p. of an engine is usually measured directly, although a knowledge of the i.h.p. obtained from the indicator diagram enables the engine-power losses to be computed.

The usual methods adopted in automobile-engine practice for measuring b.h.p. are: (1) the Prony brake, (2) the fan brake, and (3) the torque-reaction method.

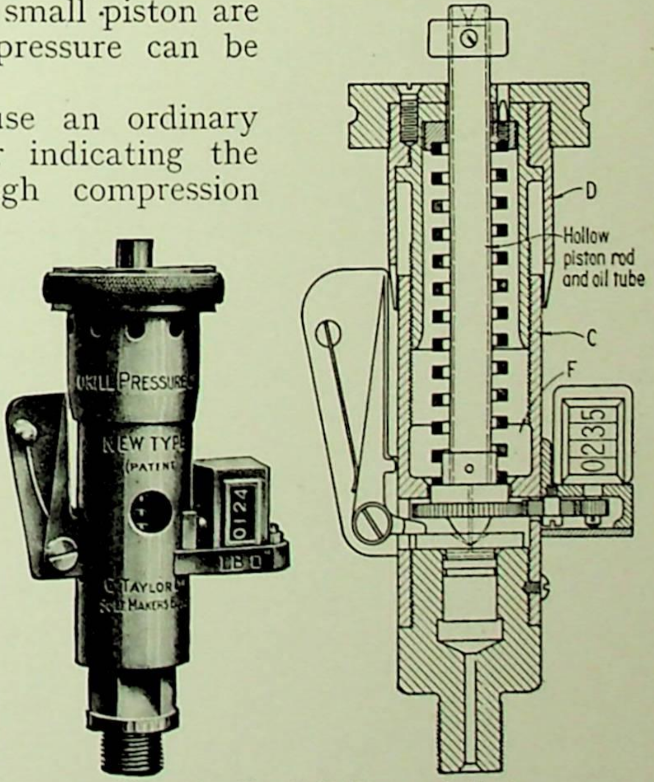


Fig. 266.—The Okill Pressure Indicator.

The Prony brake method measures b.h.p. by absorbing all the power, and consists of a brake drum, which is generally water cooled on account of the heat generated by the friction, provided with brake blocks bearing upon its surface. An arm is provided with a spring balance[1] at the outer end, and an extension of this arm carries a balance weight for neutralising the turning moment due to the unbalanced weight of the arm. This brake drum is fitted on to an extension of the crankshaft of the engine under test. When the engine shaft is running, it will be seen that the brake drum, which is keyed on to it, will rotate and will tend to drag the brake arm around with it, but owing to the constraint of the spring balance this will be prevented, and the force at the end of the brake arm will be registered on the spring balance. If this force be called W lb., and R the length in feet from the centre of the balance to the axis of the crankshaft, then the horse-power can be calculated, if the revolutions N, per minute, are known, as follows :

$$\text{b.h.p.} = \cdot 0001904\ W\ R\ N.$$

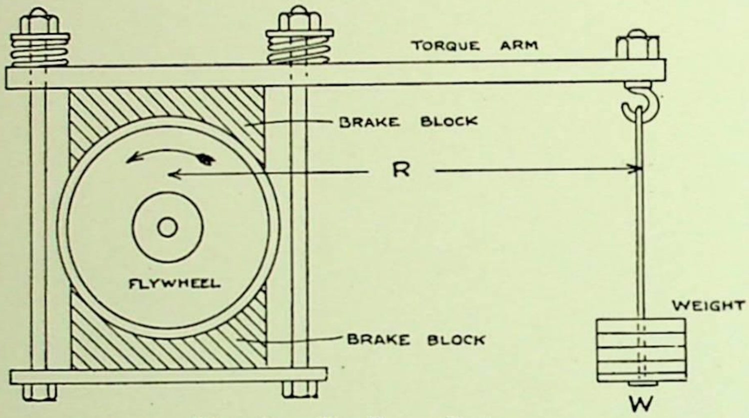


Fig. 267.—The Prony Brake.

This method is frequently modified by using a flexible belt provided with a series of wooden friction blocks, or by a belt alone, and with a spring balance on each side of the parallel portions of the belt. The difference of their readings gives the force W, and the radius of the drum the value R used in the above formula.

Another convenient method which has been widely used for testing petrol engines is the Walker fan brake one, in which a pair of flat rectangular plates of known size are mounted on opposite sides of an arm clamped at its centre to the crankshaft, as shown in Fig. 268. It is only necessary to measure the crankshaft revolutions in order to ascertain the b.h.p., for these fans have been calibrated by the manufacturers on an electric motor, and the horse-power and speed plotted on a series of curves, similar to those shown in Fig. 269. By employing a series of plates, and utilising the different radii of the arms, it is possible to test engines from low to high horse-power. Thus a pair of plates 6 × 6 inches at an overall radius of 11·6 inches will absorb about a

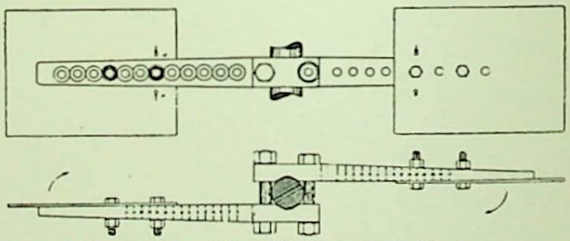


Fig. 268.—The Walker Fan Brake.

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[1] Or scale pan W as shown in Fig. 267.

quarter h.p. at 500 r.p.m., and 10 h.p. at 5,000 r.p.m. A pair of plates 17 × 8½ inches wide will absorb nearly 60 h.p. at 1,000 r.p.m., but by moving the plates inwards on the arms to the smallest radius they will absorb only 16 h.p. at the same speed. Thus a wide range of horse-powers may be measured in this manner, by a simple variation of the positions of the plates on the arms, and by using different sizes of plates. The maximum size of fan brake will absorb 150 h.p. at 1,000 r.p.m.

The third method, which is very widely used for car and aircraft engines, is the torque arm method, in which the power is absorbed by either an electric generator, the casing of which is not fixed but can rock on knife-edges, or by an hydraulic brake provided with a rocking casing ; in each case the engine power is absorbed by the generator or water brake.

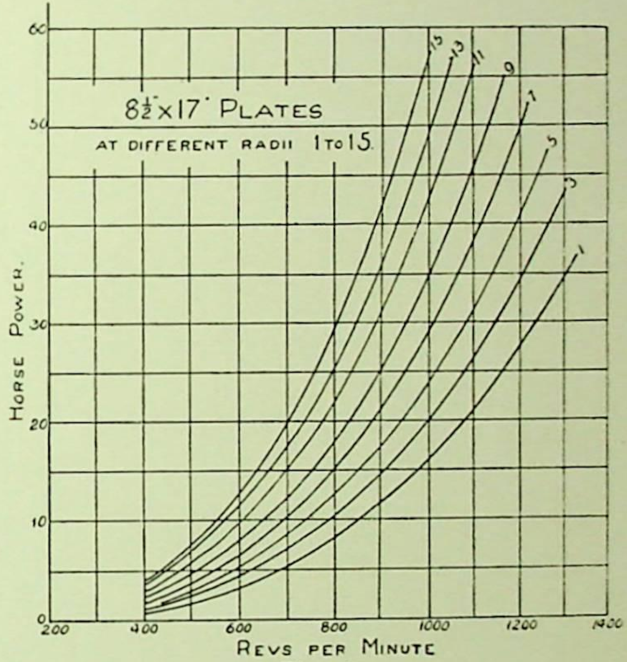


Fig. 269.

Fig. 270 illustrates the principle of this absorption dynamometer. To the shaft (which is coupled to the engine crankshaft) is fitted a rotor

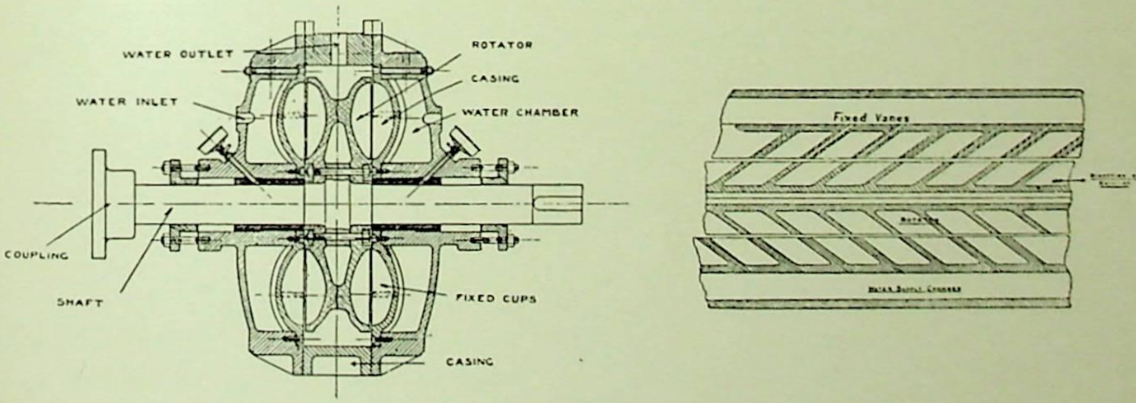


Fig. 270.—The Froude Dynamometer Principle.

provided with a series of vanes or pockets, and the internal faces of the surrounding casing are also pocketed in the same way. The outer casing carries the bearings for the shaft, with water-tight glands at each end. The whole casing is filled with water, so that when the shaft is running the water will be continuously churned up by the fixed and rotating vanes and will thus create a great resistance or torque. The outer casing is free to rock about the axis of the shaft, but is constrained from so doing by means of an arm carrying a scale pan and weights, or by a spring balance, as shown in Fig. 271. It will be seen that the turning effect on the casing (which is equal and opposite to the engine's torque which it is desired to measure) can thus be measured, and is the product of the weight on the balance arm (or spring pull) and the length of this arm. It is really the resistance offered by the water to the motion of the rotor which reacts on the casing, and tends to drag it around, this tendency being counteracted by the balance arm and weights.

If W is the load at the end of the arm, and N the engine revolutions per min., then for an arm 5 feet $3\frac{1}{10}$ inches long (or 1432·4 mm.) the formula for the power is:

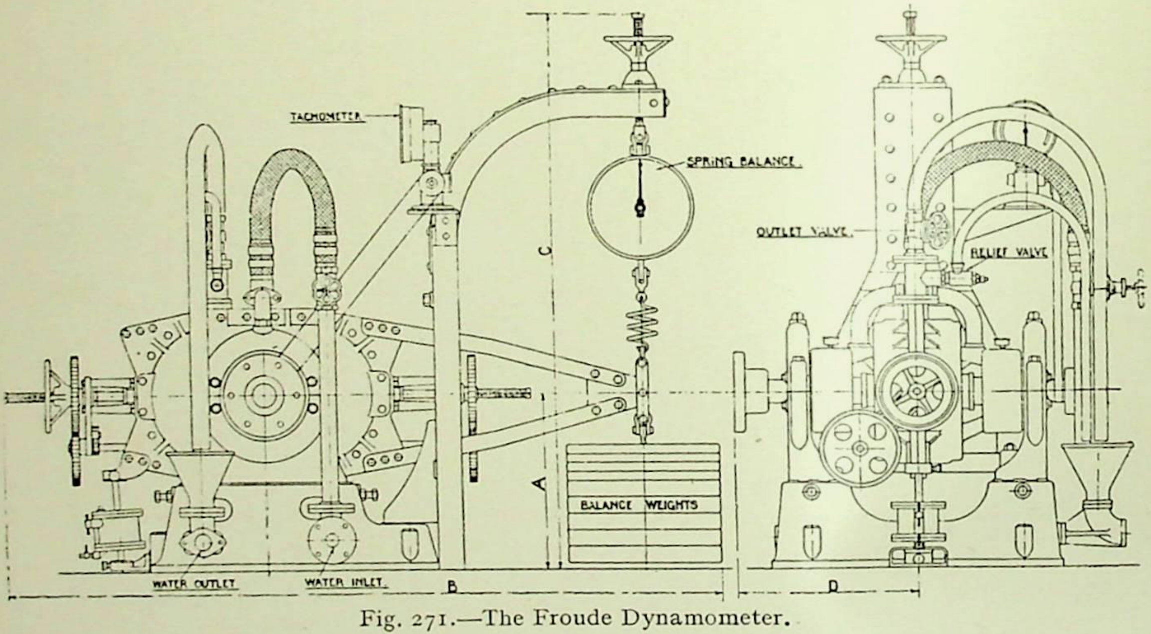


Fig. 271.—The Froude Dynamometer.

$$\text{b.h.p.} = \frac{W \times N}{1000}$$

For other lengths of arm there will be other values for the divisor; the same formula as given for the Prony brake can also be used.

There is a constant supply and ejection of water to and from the casing whilst the engine is running, in order to carry off the surplus heat caused by the resistance of the fixed and rotating vanes. The water inlets and outlets are shown in Fig. 271, together with the tachometer, on the central pillar in the l.h. view, for indicating the speed, and the spring balance and balance weights at the end of the balance arm. This dynamometer, which is made by Messrs. Heenan

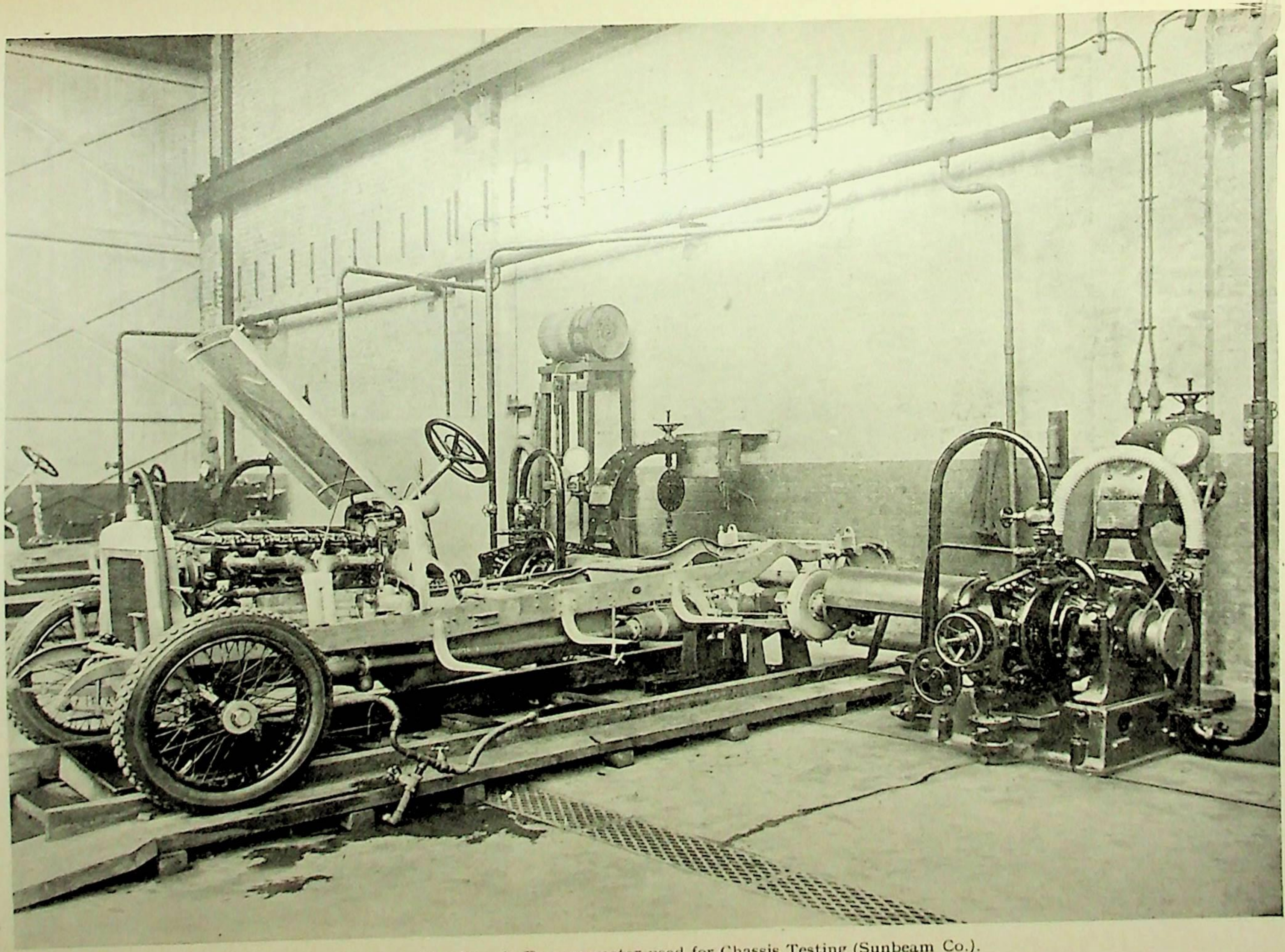


Fig. 272.—The Froude Dynamometer used for Chassis Testing (Sunbeam Co.).

and Froude, is provided with means for regulating the load at any speed, by interposing thin metal plates between the casing and the rotor, in such a way that by moving an external hand wheel these thin plates can be advanced so as to cut off communication between the rotor and casing, and so diminish the resistance of the dynamometer. The fan brake cannot be varied in load at any speed, except by fitting it in a circular casing, provided with a central entry orifice, the area of which can be varied at will.

Fig. 272 illustrates the method adopted by Messrs. The Sunbeam Motor Co. for testing the power output at the road wheels of their complete chassis, by coupling up the back axles to water brakes of the type just described. In this way it is possible to measure the efficiency of the transmission on all of the driving gears, and to test the differential gears also.

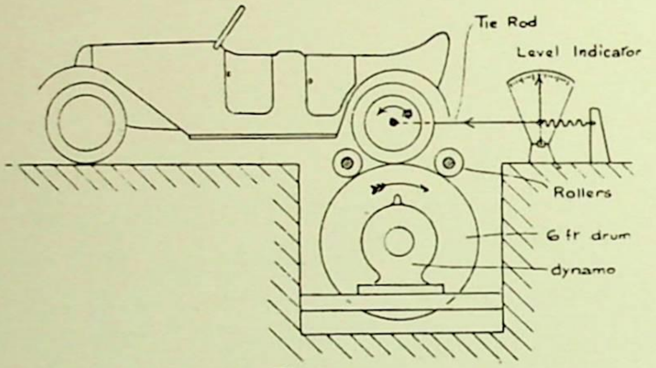


Fig. 273.

If the b.h.p. of the engine alone is measured, and be noted by H_1, and then the h.p. at the road wheels, or rear axles, by the above method, and be denoted by H_2, then the efficiency of the transmission expressed as a percentage is :

$$\text{Transmission efficiency} = 100 \,.\, \frac{H_1 - H_2}{H_1} \text{ per cent.}$$

During these brake tests the petrol consumption can be measured by feeding the fuel to the carburettor, at a fixed head, from a calibrated glass measure fitted up for the purpose. Indicator diagrams can also be taken, if a more complete investigation is to be made.

Fig. 273 illustrates an alternative method used for testing chasses, and consists of a large drum on the armature shaft of a generator, mounted in bearings. The rear wheels of the chassis are arranged to run on this drum, rollers being provided to prevent the wheels moving off the drum. The pull of the car, or traction effort, is measured on a calibrated spring and scale, or directly off a spring balance. This pull in lb., multiplied by the peripheral speed of the drum in feet per second and divided by 550, gives the h.p. at the road wheels.

CHAPTER VIII

THE CHASSIS AND FRAME

THE term "chassis" is applied to the combination of the engine, the transmission to the rear wheels (in a few cases the drive is on the front wheels), the wheels, axles, steering gear, controls, dashboard, and to the frame to which these various members are bolted or held. It will be evident, therefore, that the chassis is a complete self-contained propelling unit, and is usually quite independent of the body, or bodywork. The chassis is thus sufficiently complete, in itself, to be able to run under the power of its engine on any road or incline—indeed it is a familiar sight around large motor works to see the stripped chassis being tested on the road, the driver being accommodated on an upturned box for a seat—and is frequently sold in this condition, that is, without the body, for individuals who wish to fit their own body on the chassis.

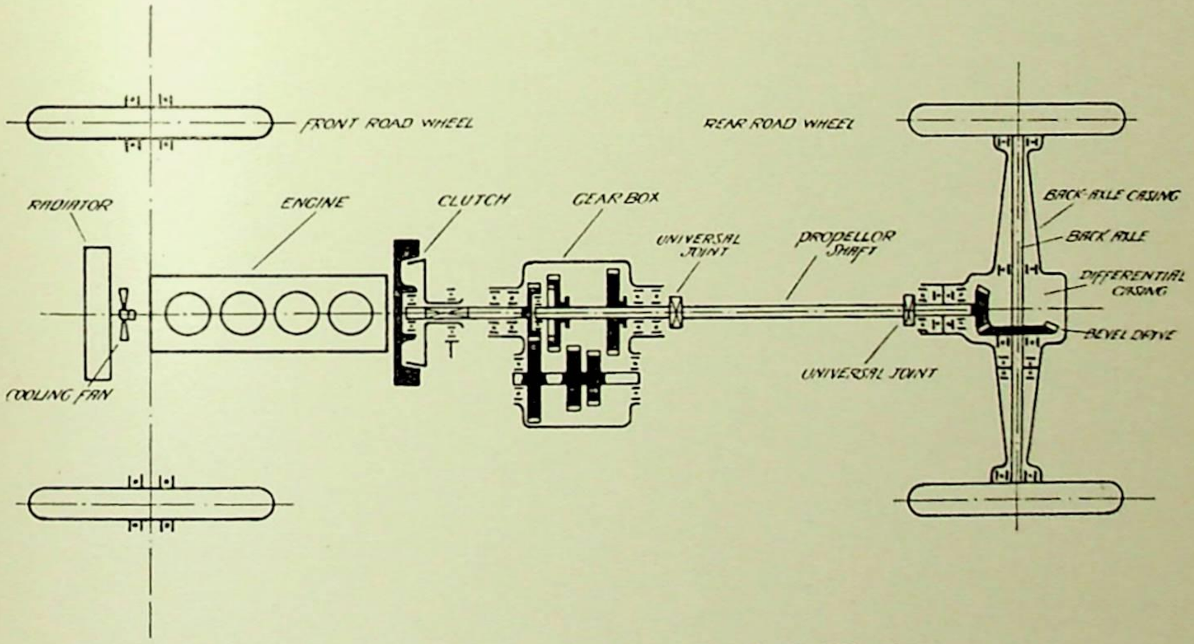


Fig. 274.—Illustrating the Layout of a Typical Chassis.

The general layout of a chassis is shown in Fig. 274, which is purely diagrammatic, however, and is given only for the purpose of illustrating the positions of the various component parts.

It will be seen that in the usual arrangement the engine is at the front end of the chassis, with the radiator and starting handle at the extreme front. This arrangement enables the engine to be readily got at, and ensures an efficient cooling system and starting arrangements; in addition the controls can easily be arranged immediately in the vicinity of the parts they operate. Thus in the case of the central gear box, bolted directly to the engine, the gear lever is in the correct position for the driver. An alternative arrangement, used in the case of the Renault cars, is that in which the radiator is behind the engine, with a fan to draw air through it; this enables a smaller sloping bonnet to be employed, and renders the engine much more accessible, as there is no radiator or water pipes, etc., to obstruct one's movements, and a bonnet can be employed which is hinged at the rear end to the dash in such a manner that it can be swung upwards from the front, and, by means of a steel stay, can be fixed so as to be quite clear of the engine.

Reverting to Fig. 274, it will be seen that the engine (a four-cylinder one in this case) is connected to the gear or *change-speed* box through the agency of a clutch, the object of which is to enable the engine to be disconnected from the gear box or road wheels during the process of changing gear, or when coasting downhill (some drivers prefer to depress the clutch and to hold it out, by means of the clutch foot pedal provided). The flywheel in most modern cars forms one of the members of the clutch, so that the latter is quite close to the engine.

In the integral system of building the engine clutch and gear box in one unit, there is usually a plain shaft between the clutch and gear box, which forms one of the driving shafts of the latter. Where the gear box is a separate unit, bolted to another cross member of the frame, it is inadvisable to utilise a solid shaft to transmit the drive from the clutch to the gear box, as there is a possibility of the engine and gear-box frame members each being slightly distorted in a different manner, which would cause a binding of the driving shaft in its bearings. Moreover, it is difficult in assembling the engine and gear box to ensure the very accurate alignment of the centres or axes of the engine and gear-box shafts on a comparatively rough chassis frame.

For this reason it is the general practice to provide flexible couplings or joints, such as the Hooke's joint (or universal) or the Hardy disc couplings, between the engine and gear box; this enables the drive to be very satisfactorily transmitted in spite of a want of alignment or rigidity in the chassis frame cross members.

The gear box, which is described in another chapter of these volumes, is simply an arrangement of gears, or chains, the ratios between which can be varied by means of sliding dog clutches so as to vary the torque or turning effort at the rear wheels. The gear box enables the rear-wheel revolutions (per minute) to be varied, relatively to the engine revolutions, in a series of steps (three, or four, according as the gear box is of the three- or four-gear type) so as to obtain a greater or lesser driving torque at the road wheels. Of course, as the torque on the wheels becomes greater, so does the driving or forward speed become less, and vice versa.

It is a definite law of mechanics that the product of the turning moment or torque by the speed (r.p.m.) is constant for a given energy supply (in this case from the engine), and that it is only possible to increase one at the cost of reducing the other in order to maintain the product constant.

This is very nearly the case in automobiles, but a slight departure results in the torque-speed relation due to the fact that the efficiency of the gears is rather lower when working on the low gears than when driving directly on top ; this difference usually means a loss in turning moment of about 5 per cent. on the low gear. A reversing-gear system is also included, for reversing the car. From the gear box the power is transmitted to the differential gear and rear axles, through a comparatively long shaft, known as the propeller or cardan shaft. This shaft, which is described in another chapter, is usually hinged, or provided with a universal joint, at its forward end, in order to enable it to swing up and down with the road wheels and back axle, under the influence or yield of the rear-springing system.

Since the rear springs generally move vertically, and the rear end of the propeller shaft moves in an arc about the gear-box end as centre, there would be a great stress on the drive unless some end movement between the propeller shaft and the axle were allowed. This is usually obtained by embodying a special type of sliding coupling near the differential, such that this end of the propeller shaft can transmit its drive, and at the same time is free to slide a little endwise relatively to the differential drive ; at the same time there is embodied a universal joint in this sliding or plunging coupling.

The power is then transmitted to the crown wheel, situated in the back-axle casing ; this wheel may be either a plain bevel wheel, a helical or herringbone bevel wheel, or a worm wheel according to the type of final drive employed.

The next components which transmit the power are the differential gears, the function of which, as has been seen earlier, is to enable the two rear wheels to run at different speeds, as when the car is turning corners, or when the resistance to one wheel is greater than that of the other ; thus if one wheel sticks in the mud whilst the other is on harder ground, the former will spin around in the mud, whilst the other remains stationary. Further, when one of the rear brakes is more effective than the other, it will frequently lock this wheel and cause it to skid whilst the other rotates normally. From the differential or balance gear the power is transmitted to the road wheels through the axle shafts which rotate in suitable bearings in the axle casing.

We have now traced out the manner in which the power (or energy) of the engine is transmitted to the rear wheels, there to be utilised as a turning effort for propelling the chassis, or car, forwards or backwards.

The arrangement described is now almost universal, and it has been very highly developed in every detail so as to be almost faultless in cars of repute. There are signs, however, of a tendency to break away from this orthodox design, and to reduce the number of parts between the engine and the driven wheels. If, for example, the gear box could be incorporated in the engine on the one side, and its drive to the wheels in the axles of the wheels on the other side, the universal joints and plunging couplings could be cut out, and a marked saving in both weight and efficiency obtained.

A few years ago vehicles, such as the Phanomobile, were employed on the road in which the engine was incorporated in the single front wheel and drove it through a speed gear and clutch. The whole front wheel, with its engine gear, could be rotated about a vertical axis, for steering purposes, similar to the steering of an ordinary tricycle.

Another type of vehicle is the three-wheeled runabout, such as the Autocarrier "Sociable" or Tradesmen's vehicle, the Morgan, Castle-Three and others, in which the drive is to the single rear wheel, either through a chain or shaft and bevel pinion drive, and in which steering is arranged in the usual four-wheel practice, by worm-and-sector, or in the cheaper vehicles by bobbin and cables. No differential is required in the case of the three-wheeled vehicle.

Having reviewed the more common systems or arrangements for transmission of the power of the engine to the wheels which convey the driving effort to the road, it is proposed to consider in more detail one or two typical motor cars, and other chasses.

Fig. 275 shows in both side elevation and plan the Hispano-Suiza chassis arrangement (as line illustrations) in order to explain the different features more clearly.

The various components have been marked in these illustrations, and to the beginner a careful study of the different components and their arrangement is recommended.

The system of springing can be clearly followed, and the manner in which the long semi-elliptic springs are fixed—pivoted at the one end and shackle-pivoted at the other—to allow for their endwise extension when flattening occurs under load or road shocks, can be seen. The side view shows the design and general shape of the channel-sectioned steel frame, with its anchorings for the ends of the springs, the radiator, engine, gear box, and propeller-shaft forward end ; the pronounced arch at the rear end of the frame is to afford vertical clearance for the back-axle movement under the spring action. Another feature which should be noticed is the manner in which the frame tapers down in vertical depth towards the two ends of the chassis. This is in accordance with the design of beams, which taper in accordance with the variation of the loads, or stresses, so as to give uniform strength and minimum weight.

The plan view shows how the frame is braced literally, by its cross

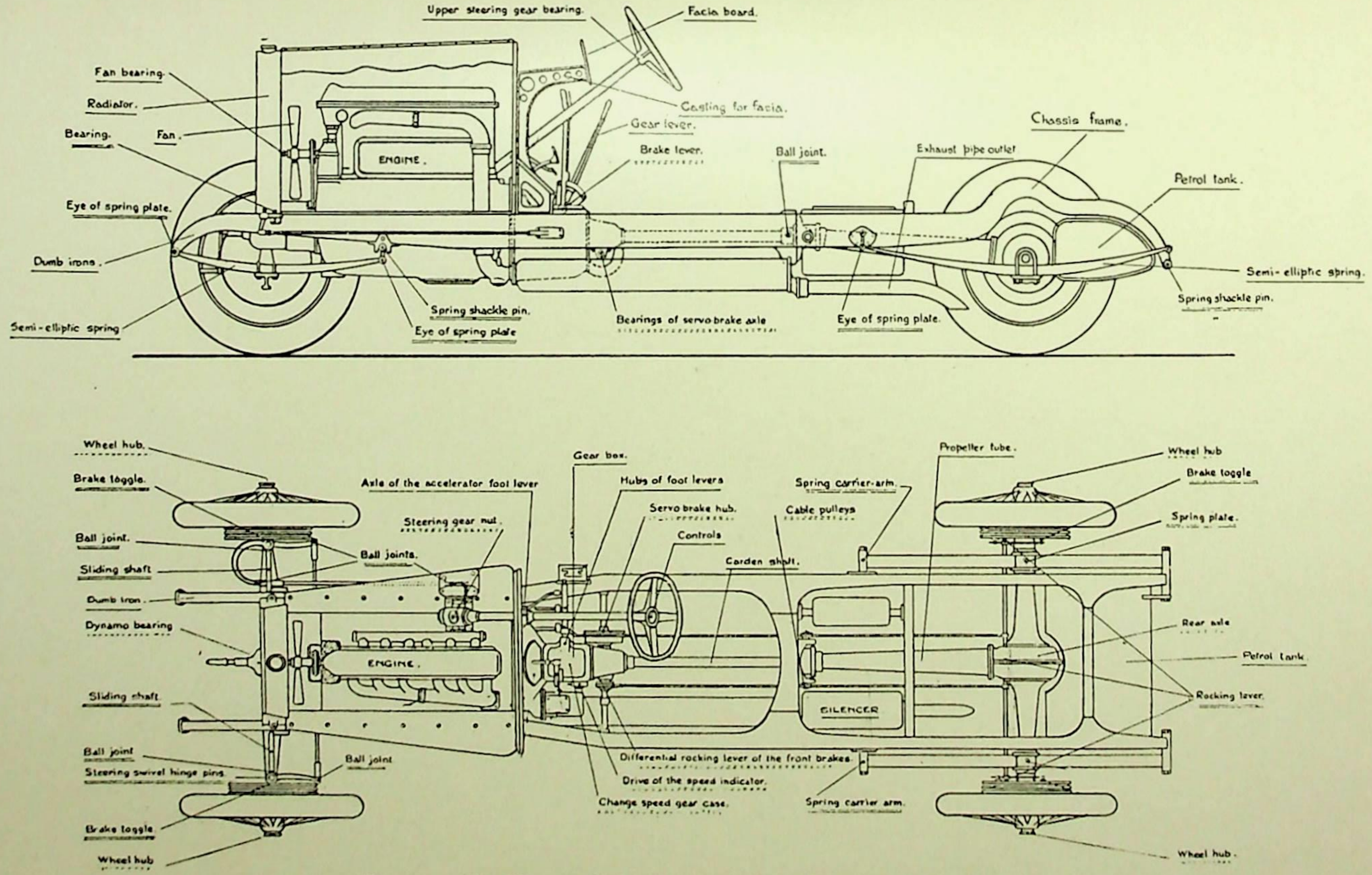


Fig. 275.—The Hispano-Suiza Chassis, in Side Elevation and in Plan.

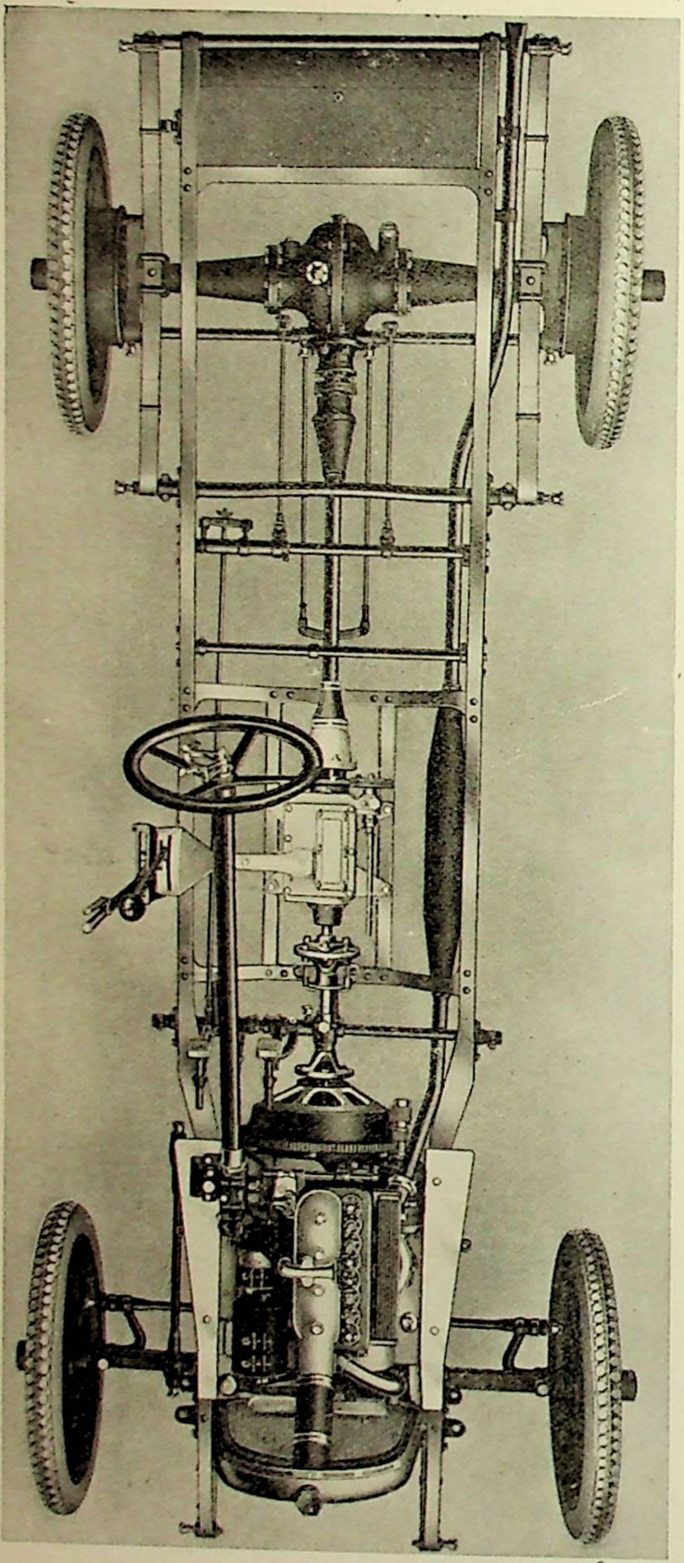


Fig. 276.—Plan View of 11·9 Bean Chassis.

members, which serve also to provide housings for the various components. The chassis invariably tapers inwards toward the front, the width there being fixed by the engine, whilst it tapers outwards at the rear, in order to provide a sufficiently wide base on which to bolt the bodywork.

The anchorage of the steering-gear column is shown in the lower view, and also the cast-aluminium bracket from the dashboard, to carry the facia or instrument panel, in a convenient position to the driver and for the steering wheel; the various control levers and pedals can be seen beneath this bracket.

The petrol tank is carried under the rear end of the chassis frame, and is shaped to suit the longitudinal members. In this case the capacity is 24 gallons, and the Autovac system of fuel feed is employed. The engine is a six-cylinder monobloc, with a bore of 100 mm. and stroke of 140 mm., and is based upon the well-known aircraft engine—the Hispano-Suiza.

Fig. 276 is a photographic illustration in plan view of the 11·9 Bean chassis, the principal dimensions of which are as follows: wheel base, 8 feet 6 inches; track, 4 feet 1 inch; length overall, 12 feet 1 inch; width overall, 5 feet 1 inch; body space, 7 feet 4 inches; weight of complete chassis 13·5 cwt.; ground clearance, 9 inches; engine, 69 mm. bore by 120 mm. stroke, giving a cubical capacity of 1,794 c.c. The frame shown is a pressed steel one, of channel section, and is amply stayed with cross members.

The rear axle is of the three-quarter floating type, and the casing is made up of four separate castings bolted together. The plan view shows clearly the arrangement of the engine and radiator, the toothed flywheel (for the starting motor pinion on the left side), the two sets of Hardy disc type flexible couplings between the engine and separate gear box, the gear box which is bolted at four points to two longitudinal bearers attached to the two frame cross members at their ends (the rivets on the frame longitudinals can be seen clearly), the tubular propeller shaft (which has a front universal joint consisting of a hardened-steel star running in phosphor-bronze bushes, and a rear universal joint of the sliding-block type with case-hardened blocks and pins), and the long semi-elliptic springs with their anchorages. The controls for the brakes can also be seen; their cross shafts are situated between the

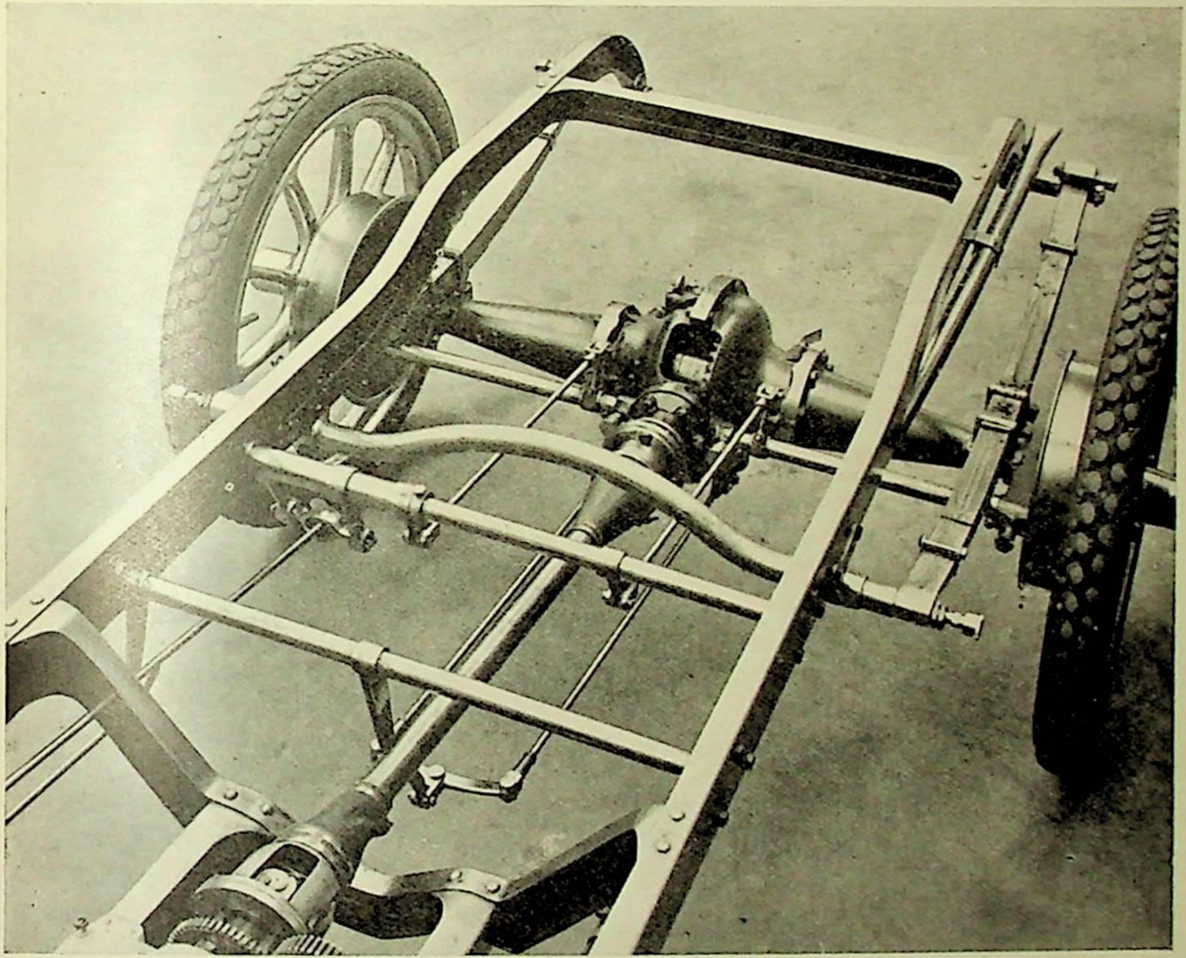


Fig. 277.—Rear View of 11·9 Bean Chassis.

gear-box cross members and the back axle. It will also be observed that the shackles and pins of the rear springs are connected, or rather are braced by means of the stout cross members of the frame, so that there is no tendency for distortion of the frame or rolling of the chassis on the springs. The exhaust pipe leads into a conical-ended silencer on the left-hand side, which is provided with a long extension pipe, provided with a flattened nozzle outlet at the extreme rear end of the chassis. The whole chassis is a typical example of a well designed and equipped English light car one.

Fig. 277 is a three-quarter plan view of the same chassis at the rear end, and shows the parts previously mentioned, in greater detail. The front universal joint, and gears for driving the speedometer off the gear box, can be seen at the front end, the cover having been removed or cut away to show the construction. Similarly a portion of the rear-axle differential casing has been cut out, in order to show the crown-wheel and differential gears. The method of securing the rear end of the gear-box channel bearers can also be seen, and it will be noticed that the channel cross member, which is of the sunk pattern, is riveted to the longitudinal members of the frame, the upper pairs of rivets being shown ; the splayed ends of these and the rear-channel cross members can also be seen.

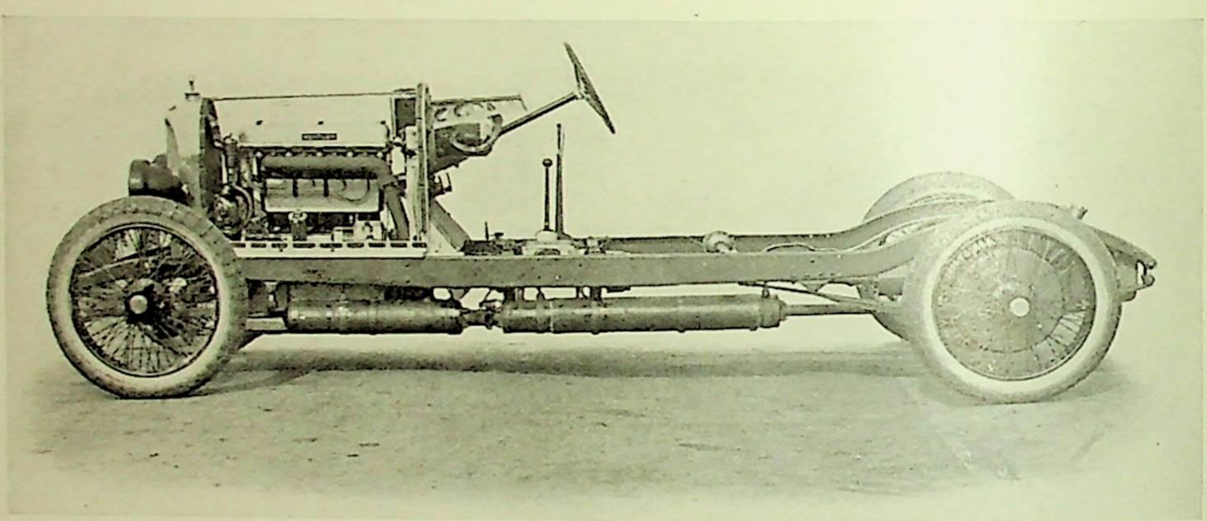


Fig. 278.—The Bentley Chassis.

The front tubular shaft is provided with bearings in the frame at its ends so that it can rock ; it carries the brake arms, which are securely locked to, and form part of, it. The arched round bar connection between the ends of the springs, with the extensions for the spring eyes and the anchorage of the rear end of the spring, can be followed. The illustration shows also the brake drums and the rear-brake lever operating shaft, situated immediately in front of the back axle.

The chassis of the 15·8 h.p. Bentley motor car, a high-grade English design, is shown in side elevation in Fig. 278; it is of low centre-of-gravity, and therefore of good stable properties on the road at high speeds. At the front end is situated the tubular cross shaft for the brake levers, operating the shorter rocking shafts and levers attached to the rear-axle casing. The springs, which are of the semi-elliptic type, are, in this case, anchored directly to the under side of the longitudinal frame members by means of the brackets shown. The back axle is of the one-piece construction. The petrol tank, with its filling cap and the petrol-pipe feed to the carburettor, is shown in this illustration. The wheel base is 9 feet 9½ inches, the wheel track 4 feet 8 inches, the overall length 13 feet 3 inches, and the road clearance 7⅝ inches.

An example of a very original design of luxurious car chassis is that of the Leyland eight-cylinder engine car, shown in Fig. 279. This chassis is made in three wheel-base lengths, namely, 10 feet 6 inches, 11 feet 9 inches, and 12 feet 6 inches, to suit different lengths and types of body. Reference will be made in a later chapter to the engine and transmission of this car.

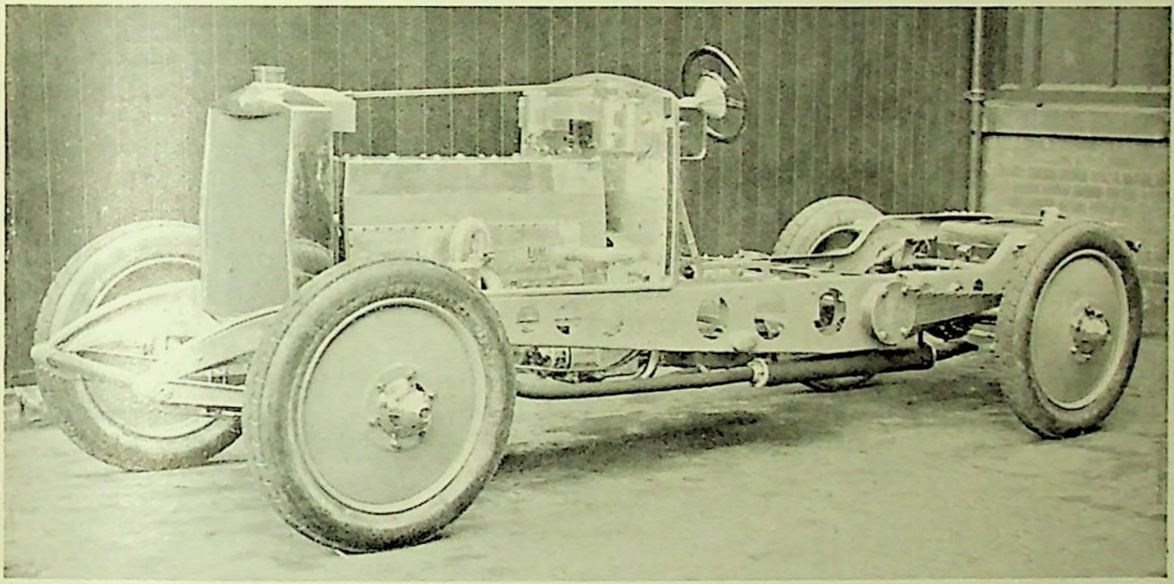


Fig. 279.—The Leyland "Eight" Chassis.

The very deep proportions of the frame longitudinals will be apparent from the illustration, these being drilled for lightness. The housing for the rear cantilever type springs on the frame, and the rigid design of the dumb irons with their cross-connecting bar and levers, are also apparent from the illustration. The front suspension consists of a pair of semi-elliptic laminated springs on rollers at either end. Each of these is anchored from the dumb irons by a torque arm, which communicates through a torsion shaft to the torque arm on the opposite side, and in this manner forms the front anti-rolling device. The rear suspension embodies a pair of quarter-elliptic springs combined with an additional torsion shaft on the spring housing shown, which absorbs the minor shocks of the road. An anti-rolling device is fitted at the rear, and Houdaille shock absorbers are fitted on the back axle. The dashboard is of cast aluminium and serves as a housing for various accessories and controls.

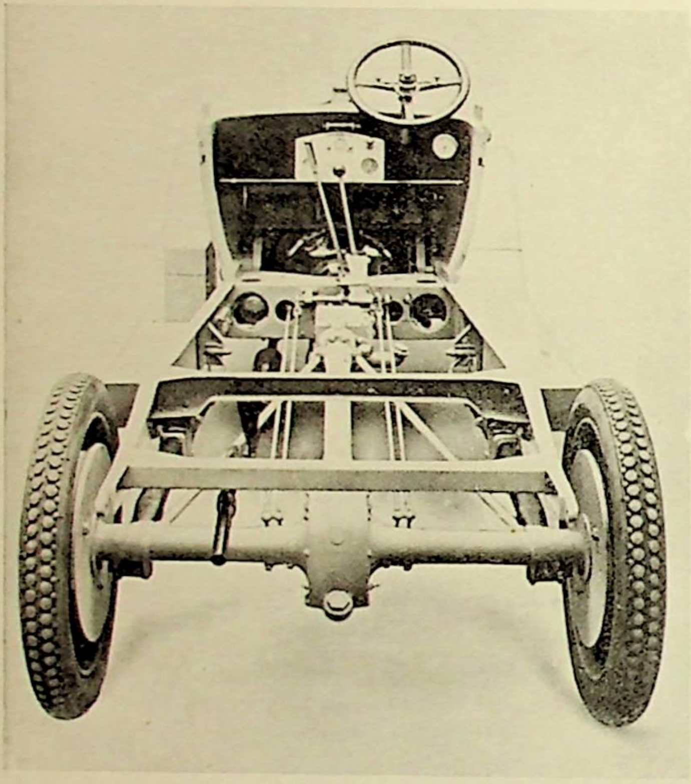


Fig. 280.—Rear View of 18 h.p Armstrong-Siddeley Chassis.

A rear view of the 18 h.p. six-cylinder Armstrong-Siddeley chassis is shown in Fig. 280. The wheel base is 10 feet, and the wheel track 4 feet 3 inches. The overall length measures 13 feet 9 inches, and the overall width 5 feet 2 inches.

The engine is of the six-cylinder vertical type, with a bore and stroke of 2·73 inches and 4·125 inches respectively.

The frame has unusually deep side members, to which the pressed-steel running boards and wing ends are secured in such a way as to make them integral with the frame. This design, with its channel-section cross members, provides an extremely rigid frame of low weight and has the effect of preventing the twisting and straining of the bodywork of the car, which otherwise would cause creaks and rattle due to frame distortion on very uneven road surfaces.

The springing in this case is by means of long semi-elliptic springs at the front and full cantilevers behind, the latter being individually adjustable to the load in a simple manner, as explained in the chapter on Suspension. The large-diameter brake drums, giving ample braking surface and long wear, and also the triangular-radius rods for guiding the rear axle, can be seen in the illustration.

SPECIAL TYPES

We have now considered the general disposition of the various components of a modern car, but before passing on to a consideration of the question of the various types of frame, to which our treatment of this subject naturally leads, a brief reference will be made to an alternative arrangement of the chassis components, namely that adopted in the Rumpler and North Lucas cars, in which the engine is behind the driver and to the rear of the chassis.

The Rumpler arrangement is shown diagrammatically in Fig. 281; the seating accommodation and general shape of the body are also apparent from the plate. If properly designed and carried out, such an arrangement will not only result in an appreciable saving of weight, and also in a number of parts of the transmission, but will give a much better bodywork and seating disposition. The true streamline ideal cannot be approached in present car designs owing to the fact that the narrowest part, namely the engine, is in front, whilst the widest part, namely the seats, is behind. The radiator would normally be behind in this car, either on the sides or transversely, with fan or induced draught cooling, so that no appreciable difficulties in connection with the cooling would result in placing the engine behind, whilst it has been shown that the engine can be made just as accessible.

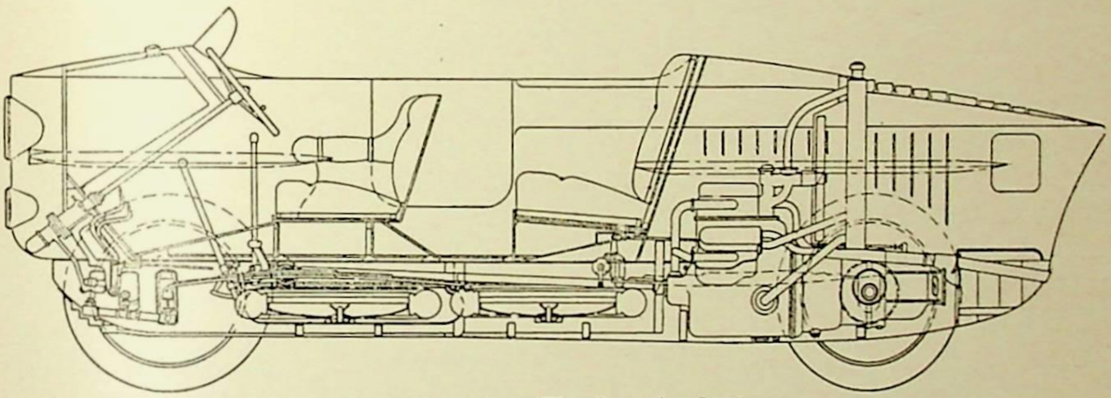


Fig. 281.—The Rumpler Car.[1]

In the case of the Rumpler car, the wheel base is 9 feet 6 inches, the wheel track 4 feet 5½ inches, and the maximum body width 4 feet 6 inches; five people in all can be accommodated with comfort. The overall length, which includes the pointed tail, is 15 feet; this is appreciably longer than usual. The car is driven by a six-cylinder engine of 74 mm. bore and 100 mm. stroke, giving a capacity of 2,580 c.c., the nominal h.p. being 20, although on the brake as much as 36 b.h.p. has been obtained.

This car has a pressed-steel frame about $\frac{3}{32}$ inch thick, and the side members follow the contour of the body; the depth of the parallel portion of the car is 12 inches, and this extends along the major portion of the frame. Flange slots and holes are provided in the vertical webs of the frame for the axles, springs, and other transverse parts.

In addition there are two large openings in the centre, which are flanged, for stiffening purposes, which give access to the spare-wheel carriers. There are five main cross members to the frame.

---

[1] Courtesy of *The Automobile Engineer*.

Springing is by full cantilevers at both front and back.

A multiple-disc clutch transmits the power from the engine through a three-speed gear box to a double set of bevel gears, one pair of which drives each wheel. The differential is on the driving shaft, upon which, also, is the brake drum.

The whole car is novel and very original, and possesses some undeniable advantages over current designs.

A car of somewhat similar arrangement is the North Lucas five-cylinder radial-engined car ; this is much smaller, however, and the air-cooled engine is set with its axis vertical, the cylinders, of course, being horizontal. In this case the framework is of the deep aluminium girder type extending across the whole width of the body so as to form the running boards. The vertical webs are 6 inches deep, and at the rear is a vertical cross member made of $\frac{3}{16}$-inch aluminium plate, and to which the power unit and transmission are bolted.

The engine is 70 mm. bore by 75 mm. stroke, and is rated at 15 h.p. ; an illustration of the engine is shown in Fig. 150.

COMMERCIAL CHASSIS

The arrangement of a commercial vehicle chassis is shown in Fig. 282, which shows the Sheffield Simplex 30-cwt. lorry. In general layout it resembles the chasses already described, the main differences being the sturdy and larger proportions, solid tyres, and different disposition of the controls to suit the commercial bodies employed. The wheel base is 11 feet and the wheel track 5 feet, there being very much more overhang of the frame at the rear in order to accommodate the lorry bodies fitted ; moreover the engine and radiator are more compact and are placed well forward. The frame itself is of simple construction, the taper being gradual and the top flat in side elevation. In plan the two longitudinals are parallel and the cross members of straight form, except for the diagonal cross bracing at the rear end. At the front end the dumb irons bend sharply downwards to take the front spring eyes ; there is also a pressed-steel cross member here, to carry the radiator, a trunnion form of radiator mounting being employed ; the front end of the engine is also on a trunnion fixture. There is a second channelled cross member close to the flywheel which carries the rear end of the engine. The gear box, it will be seen, is mounted on two other cross members, whilst there is a fifth member fitted near to the rear-spring brackets, and finally a sixth cross member across the extreme end of the frame. The semi-elliptic springs at the rear are connected at their ends by stout tubular members to take the torque of the springs. The engine employed is a four-cylinder one of 100 mm. bore and 120 mm. stroke, giving a capacity of 3,769·9 c.c. and a nominal h.p. of 24·7 ; actually it develops 28·5 b.h.p. at 1,400 r.p.m. An external leather cone clutch, three-speed (and reverse) gear box giving much lower ratios than in car practice,

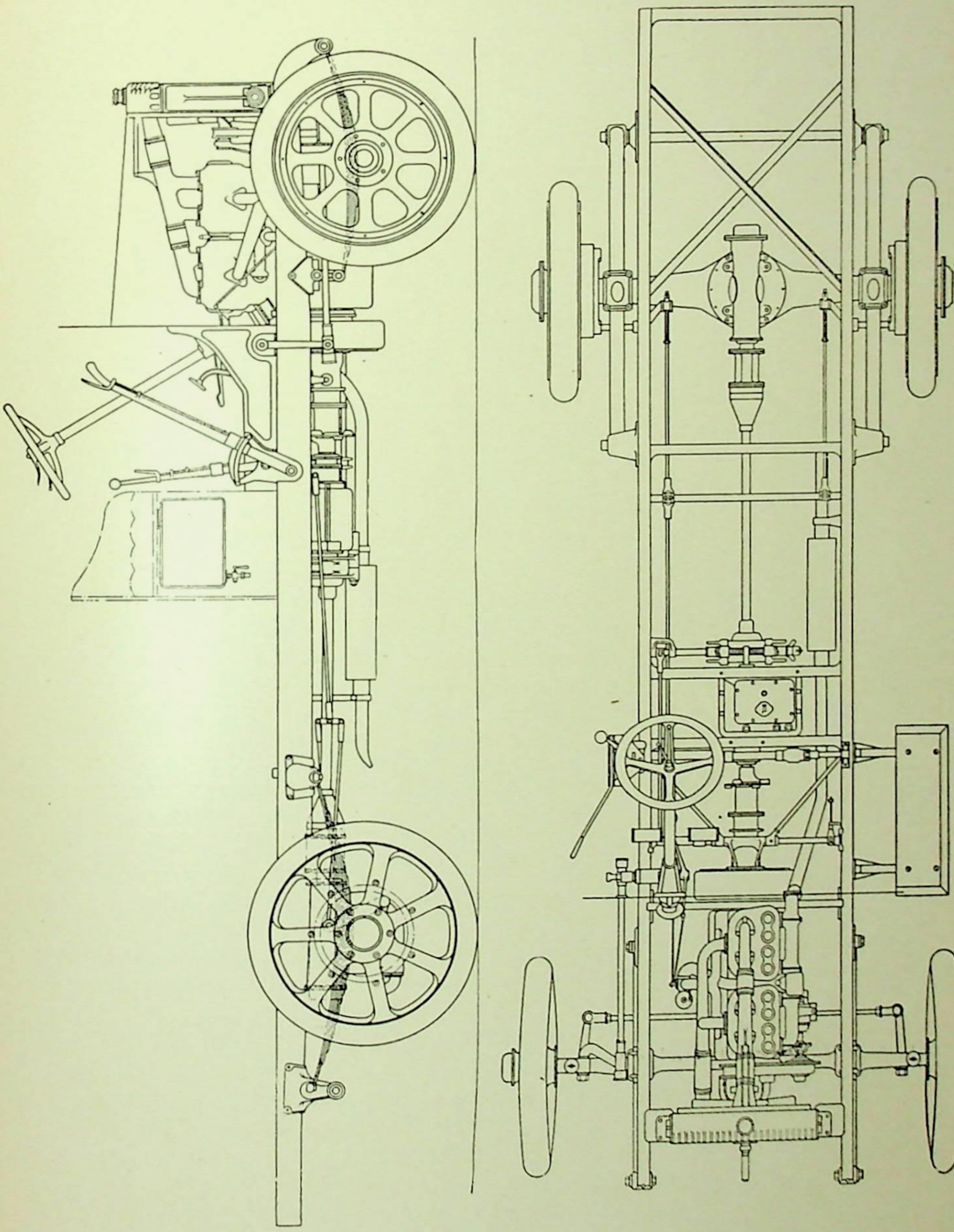


Fig. 282.—The Sheffield Simplex 30-cwt. Lorry (*A.E.*)

namely, 23·3 : 1, 12·6 : 1, and 6·6 : 1 forward, and 20 : 1 on reverse, transmission brake on the back end of gear box, the usual propeller shaft (but of solid construction), and a final drive through overhead worm and worm wheel, with full-floating splined axles carried on two sets of Timken bearings at the wheel end, are the chief features in the arrangement of this chassis.

The subject of other light-car and cycle-car chassis arrangements, and also motor-bicycle frames will be dealt with under their respective headings.

FRAMES

The general purpose which the frame fulfils will now be evident, from the somewhat frequent allusions made in the preceding pages. It will have been seen that the main object of the frame is to accommodate, or house, the engine, radiator, gear box, and steering members, to provide end fixings for the springing system, since the back and front axles with their wheels are "insulated" from the chassis frame with its members, and the bodywork which is carried on the frame.

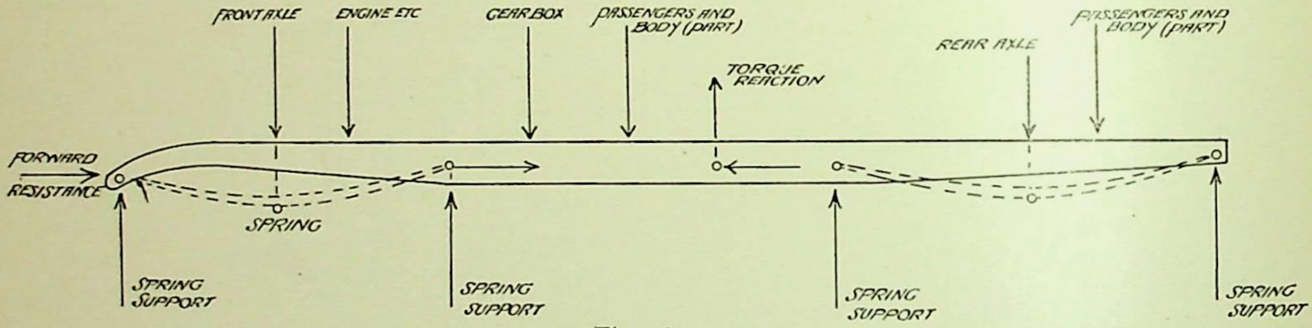


Fig. 283.

The frame also serves the subsidiary purposes of providing a more or less rigid fixing for the dashboard, brake-control shafts, the gear levers, steering-arm bracket, running boards, battery compartment, and silencer brackets.

The manner in which the various loads are distributed along the frame will determine the stresses which come on it, and will enable its proportions to be fixed, or rather checked, since there are certain practical considerations which have to be taken into account.

Fig. 283 illustrates diagrammatically how the various forces act in the case of an automobile chassis frame; by a suitable estimation of the distribution of, and a knowledge of, the actual weights of the complete car, it is a straightforward matter for the designer to check the strength of the frame all along.

Frames are usually constructed in four principal ways, namely:

(1) The pressed-steel method, which is almost universal at the present moment.

(2) The built-up tubular method, or bicycle-tube construction, which is used only upon cycle cars.

(3) The reinforced wood method, which utilises beams of straight-grained ash for the longitudinals, mortised and tenoned to the wooden transverse members, the joints being stiffened with steel plates. This method has now fallen out of use, for it was found that the members became bulky and inconvenient as the car sizes and speeds grew. These wooden frames were very resilient and gave good results with light cars, but as the speeds and weights increased, and the springing systems became more effective, it was found that the pressed-steel frame possessed certain advantages in the matter of permanent rigidity (wood being an organic substance and liable to vary) and cheapness which finally led to their adoption; it is therefore not proposed to consider the wooden frames any further here.

(4) The combined frame-and-body-member system as used in the Lancia light car and others. This method will be referred to again later.

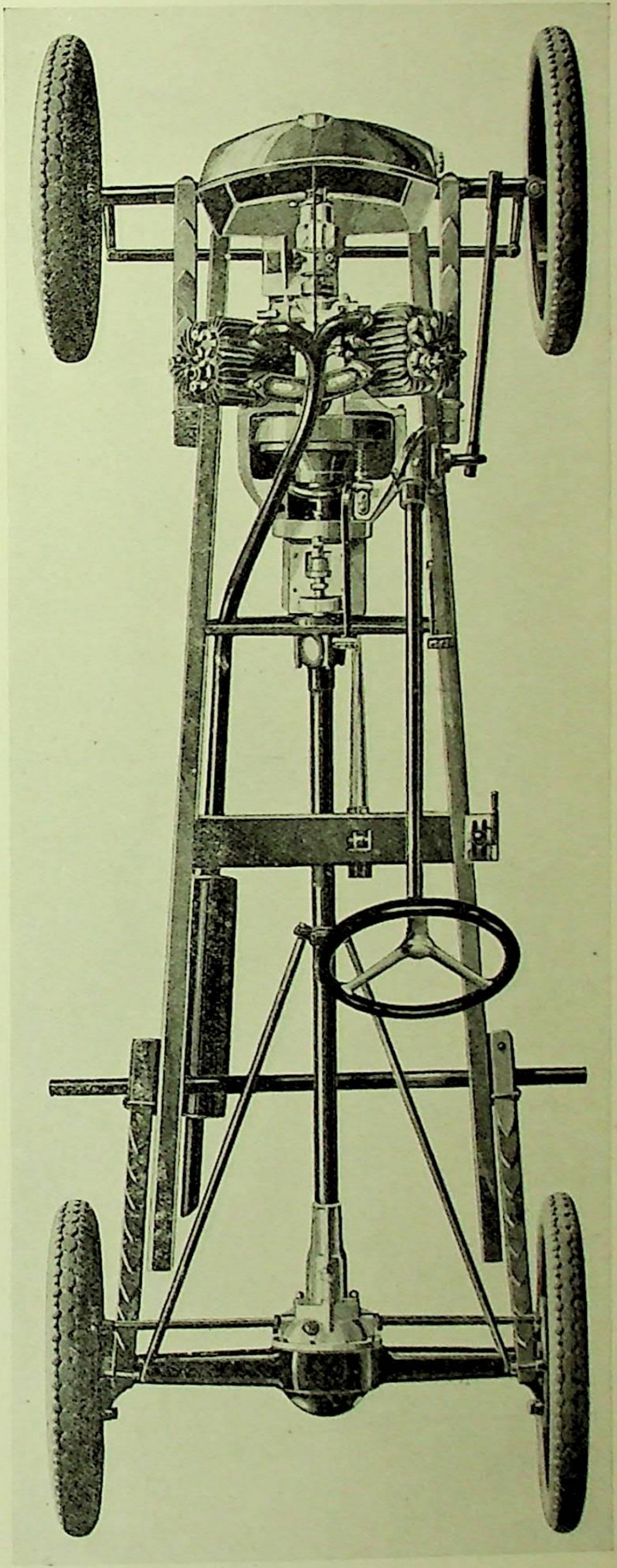
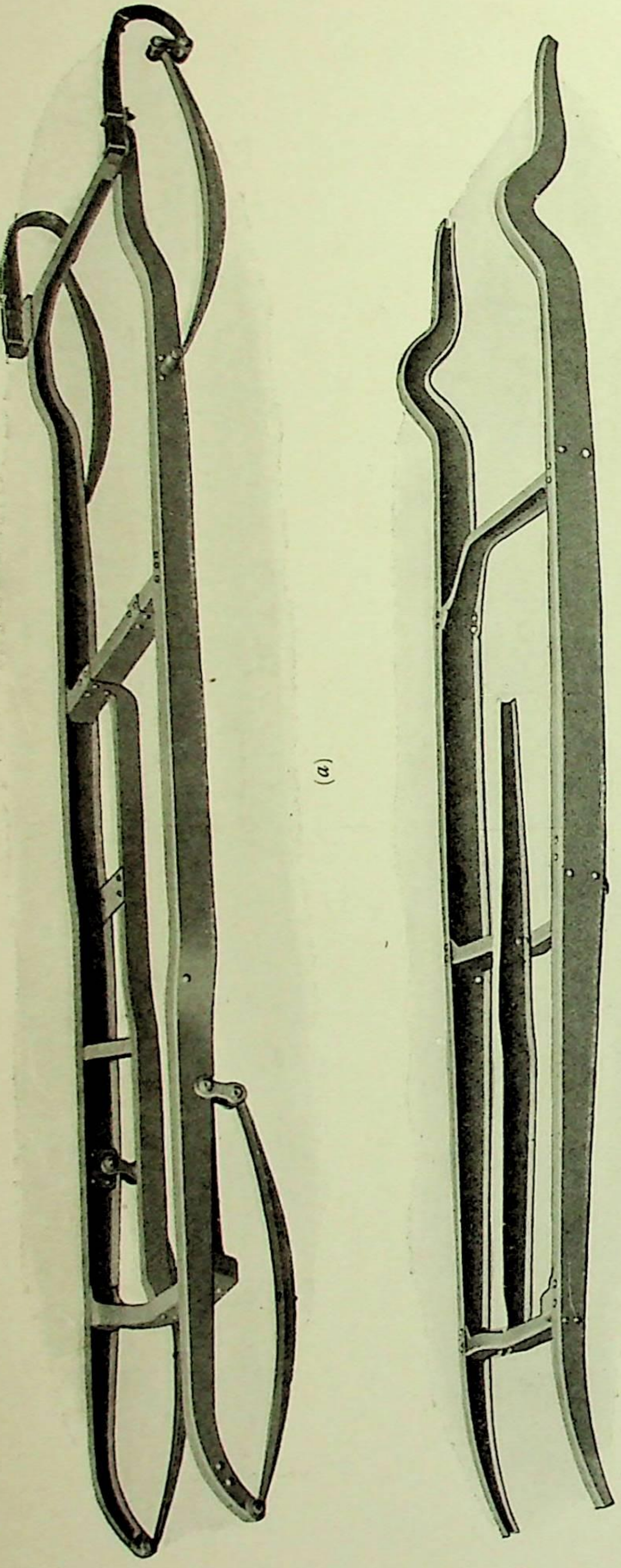


Fig. 284.—The G.N. Chassis in Plan.



(a) (b)

Fig. 285.

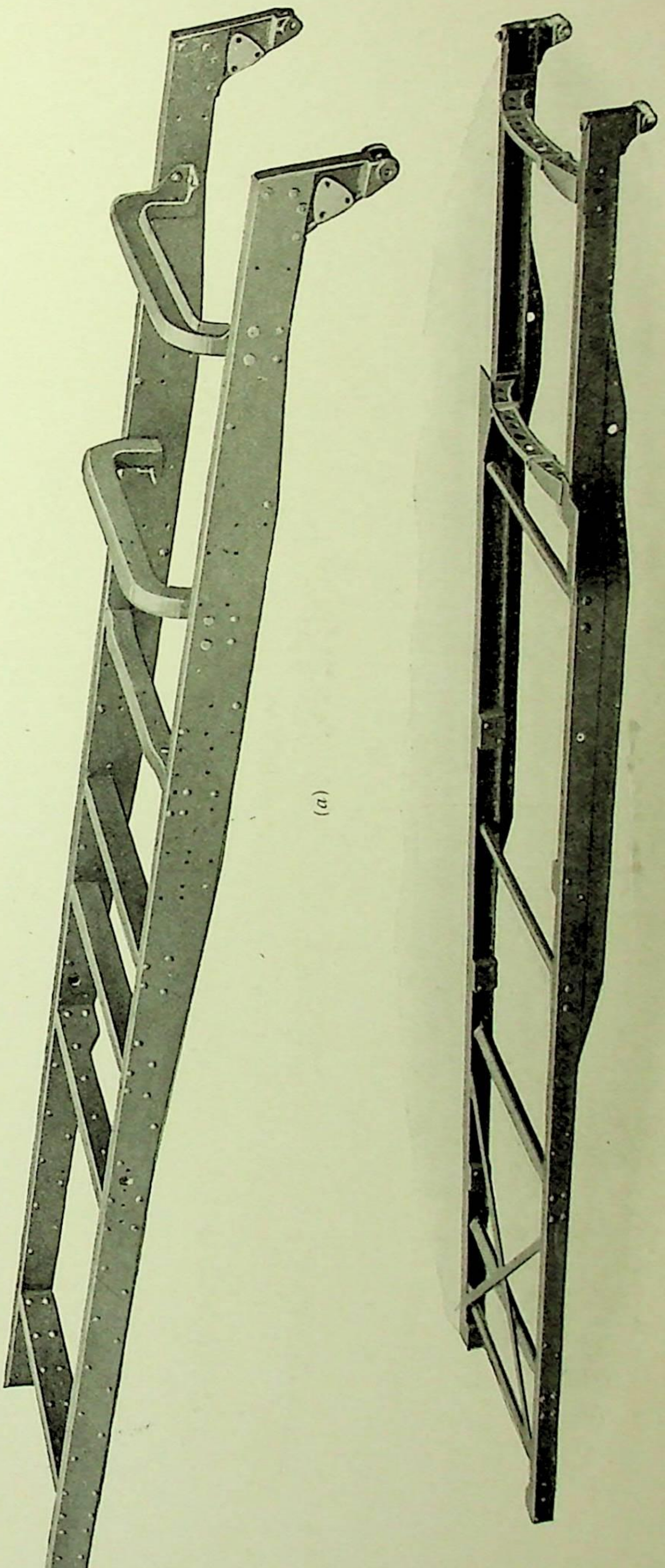
The design of steel frames is very varied, as will be seen from the illustrations given, and from the fact that frame manufacturers make many different patterns.

The simplest frame is the plain, parallel-section channel type employed upon cycle cars and certain light cars. The two longitudinals taper outwards from front to rear, and are connected by transverse members to carry the radiator, engine, and gear box, and to connect the rear ends.

Fig. 284 illustrates a typical frame of this type for a light car having quarter-elliptic, or cantilever springs as shown. It will be observed that there is no rear cross member, but that the rear springs are anchored to the transverse tube shown, which is also extended so as to carry the footboards.

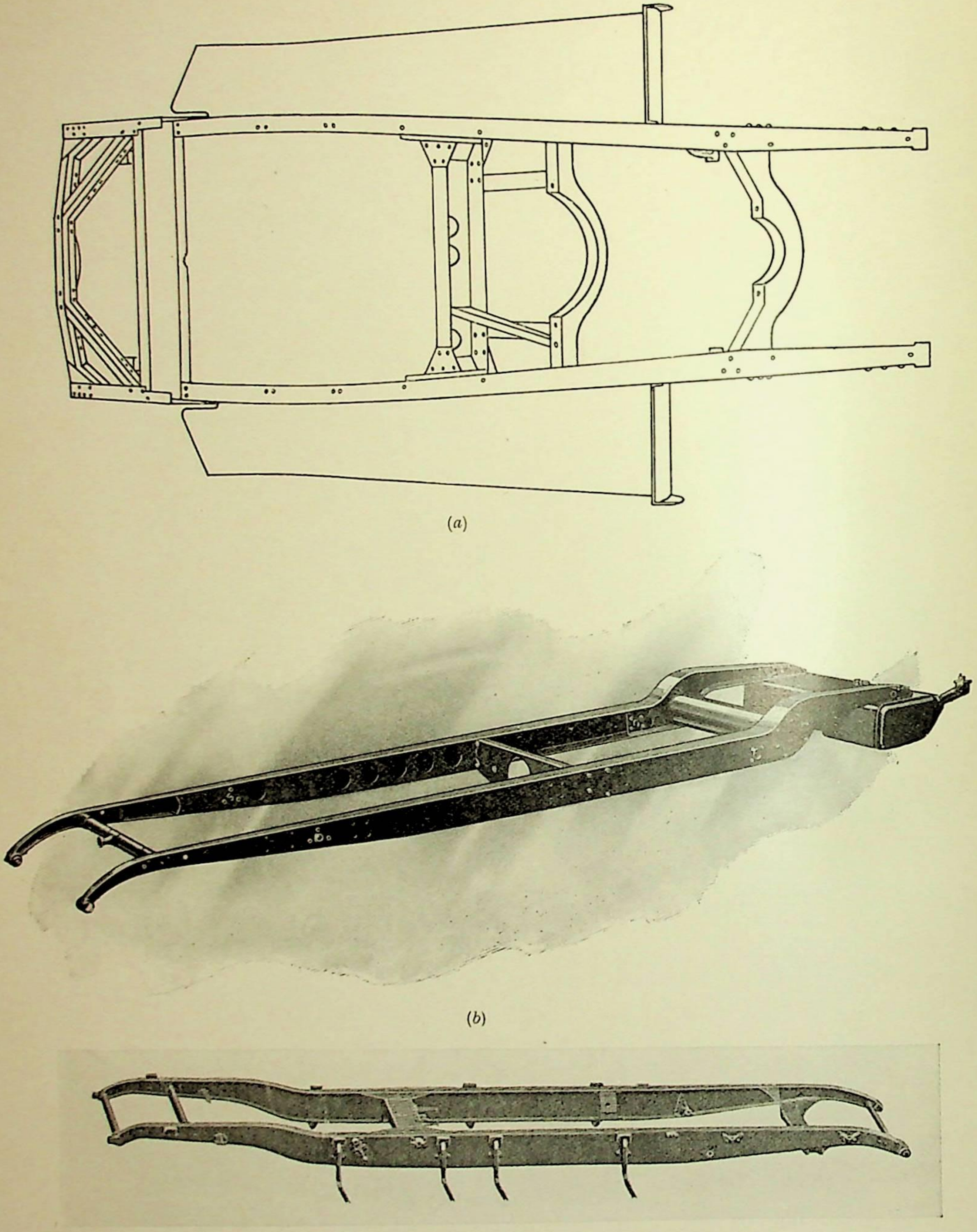
A simple form of motor-car frame made by Messrs. Rubery Owen is shown in Fig. 285 (*a*), with the semi-elliptic springs in position at the front end, and the three-quarter elliptic springs at the rear. The front transverse member is of channel section and carries the radiator, whilst the engine is bolted on the subsidiary longitudinal bearers shown. The frame is slightly upswept at the back, in order to provide clearance for the back axle as it moves upwards.

Another design of frame, but with a more pronounced upsweep at the rear end, is that shown in Fig. 285(*b*). Here the engine bearers are overhung and carry the separate gear box, universal couplings being employed. The overall dimensions of this frame are 11 feet long by 2 feet 4 inches wide at front, and 3 feet 6 inches wide at rear.


(*a*)

(*b*)

Fig. 286.



(a)

(b)

(c)

Fig. 287.

An exceedingly simple type of frame is that illustrated in Fig. 284 and employed on light cars. There are here only two transverse members, one near either end; the overall dimensions of such a frame would be about 7 to 8 feet long by 2 to 3 feet (rear) wide, the maximum channel section measuring 2¾ inches by 1¼ inch by ⅛ inch thick.

Two alternative designs of motor-lorry frames are shown in Fig. 286 (*a*) and (*b*); these are made by Messrs. Rubery Owen, of Darlaston.

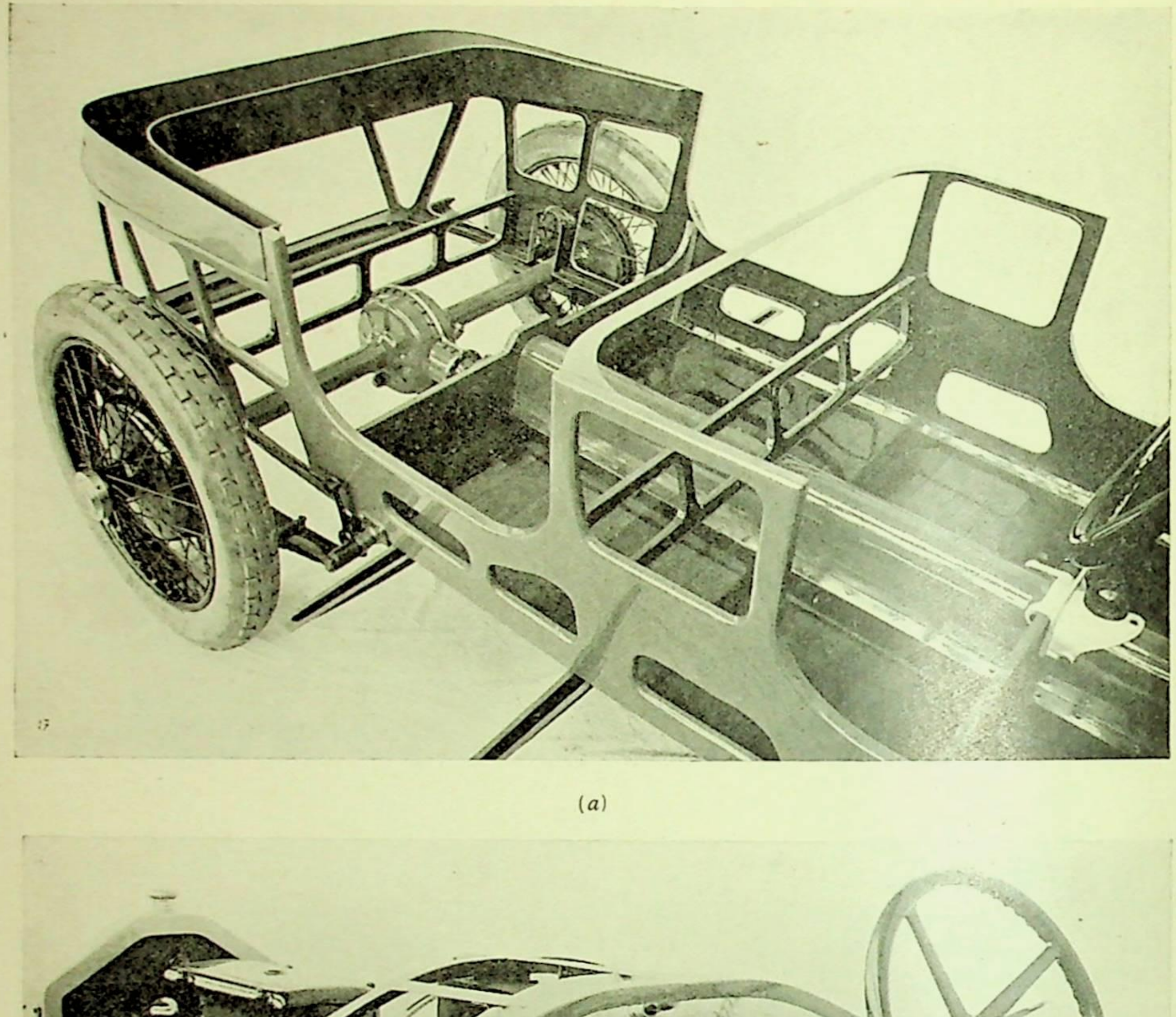
It will be observed that in both cases, as in the example shown in Fig. 286, the top flanges of the longitudinals are straight, the tapering being obtained on the lower flanges of these members. The built-up dumb irons in each case enable a cheaper form of manufacturing process to be employed. The example shown in (*a*) is of box section, whereas that given in (*b*) is of channel section with vertical web extensions.

The dimensions of a typical lorry frame are as follows: length overall, 21 feet; width overall, 4 feet 6 inches at rear, 3 feet at front; dimensions of channel at maximum section, depth 6½ inches, width of flanges 2 inches, thickness of metal ¼ inch to $\frac{5}{16}$ inch.

A more complex design of chassis frame is that shown in Fig. 287 (*a*), in part perspective plan view, namely for the 34 h.p. American Marmon car, of the six-cylinder vertical type. The special features of this frame are the engine bearers (shown in perspective), with their circular recesses for clearing the engine sump; the bolt holes for the four-engine bearer arm fixture are indicated. The built-up cross member in the centre is very rigid and carries both the central gear box and also the brackets for the control-pedal fulcrums. The reinforced rear portion accommodates both the body extension and the petrol tank and spring socket brackets. The two longitudinals are of quite simple design and taper outwards towards the rear.

The Lanchester chassis frame shown in Fig. 287 (*b*) is an excellent example of high grade car design, the proportioning of the members being carefully carried out. There is a tubular cross member at the front, so that the maximum stiffness is obtained; this member houses the radiator. The large tubular cross member towards the rear is for relieving the frame of torsional stresses in the vicinity of the cantilever springs and to serve as a rigid transverse member. The petrol tank is secured to the rear end of the frame. In connection with the longitudinal members, it will be observed that these are stiffened towards the front end by internal channels, with lightning holes in the webs, so that a strong box section is obtained.

Fig. 287 (*c*) shows the Cadillac frame, which has a maximum depth of 7 inches, and is provided with five cross members. The longitudinals have long deep (7 inches) channel section and wide top flanges. There is a gradual reduction of section towards each end with a moderate up-sweep at the rear. The central cross member to which the forward end of the torsion arm is attached is securely riveted to the longitudinals by both web and flanges, the joints being reinforced with gusset plates.



(*a*)

(*b*)

Fig. 288.—The Lancia Combined Chassis and Body Frame.

The four brackets shown on the front longitudinal member are for the running-board fixings.

The material used for chassis frames is usually a low nickel steel (2 to 3 per cent. nickel), having a breaking strength of about 35 to 40 tons per sq. inch, or a good mild steel, having a breaking strength of 28 to 35 tons per sq. inch. Nickel chrome steel, of about the same strength as nickel steel, has also been used on account of its resistance to shocks, the nickel, chromium, and carbon contents being about 3·0, 0·5, and 0·25 per cent. respectively. No heat treatment beyond normalising is given to chassis frames.

It was mentioned earlier in this section that the body frame and the chassis frame were sometimes made in common. This method has advantages from the production and also from the rigidity standpoints; but against this must be set the additional difficulties of manufacture and repair, unless steel stampings of ample proportions are used.

An excellent example of this unit construction is shown in Fig. 288 (*a*) and (*b*), in the case of the Lancia " Lambda " light car. Here, the side members are of pressed steel of 2 mm. thickness suitably lightened and flanged, with a series of riveted cross members, also made of lightened pressings, which serve to carry the tubular supports (longitudinal) to which the engine is bolted, the dash- and facia-boards, the front footboards, rear transverse members, and the metal well in which the hood and its frame drops and is concealed.

There is a longitudinal inverted U-shaped tunnel made of sheet metal, which encloses the transmission from the engine to the back axle; the running-board brackets, which can be seen in Fig. 288 (*a*), are of pressed steel U-section with stiffening flanges. The slots, with flanged edges for stiffening purposes, to allow for the vertical movement of the springs and rear axle, are also shown. Altogether this design, for a four-seater touring body, is very neat and light, and is an agreeable departure from current practice. The engine of this car has been referred to previously (see p. 125). It is of the 13·9 h.p. rated type, but gives more than 45 h.p. on the brake. The wheel base of this car is 10 feet 2 inches, the track 3 feet 7 inches, and the total weight of the chassis and body, with tanks filled, spares, tools, etc., is only 15 cwt.; this is a very low weight for this type of car, as it is equivalent to 1 cwt. for every 3 b.h.p., at the maximum output. The result of this is reflected in the high maximum road speed of this car, namely, 70 to 75 m.p.h.

The four-seater body can be provided with a detachable top so as to make an enclosed limousine type of car.

LUBRICATION OF THE CHASSIS

If the chassis be considered as a piece of working mechanism, containing the power unit, the transmission, controls, steering gear, suspension system, and wheels and bearings, it will be evident that there is quite a large number of working parts requiring attention.

The lubrication of the engine and transmission will not be considered here, but rather those components and parts which are dealt with in the present section.

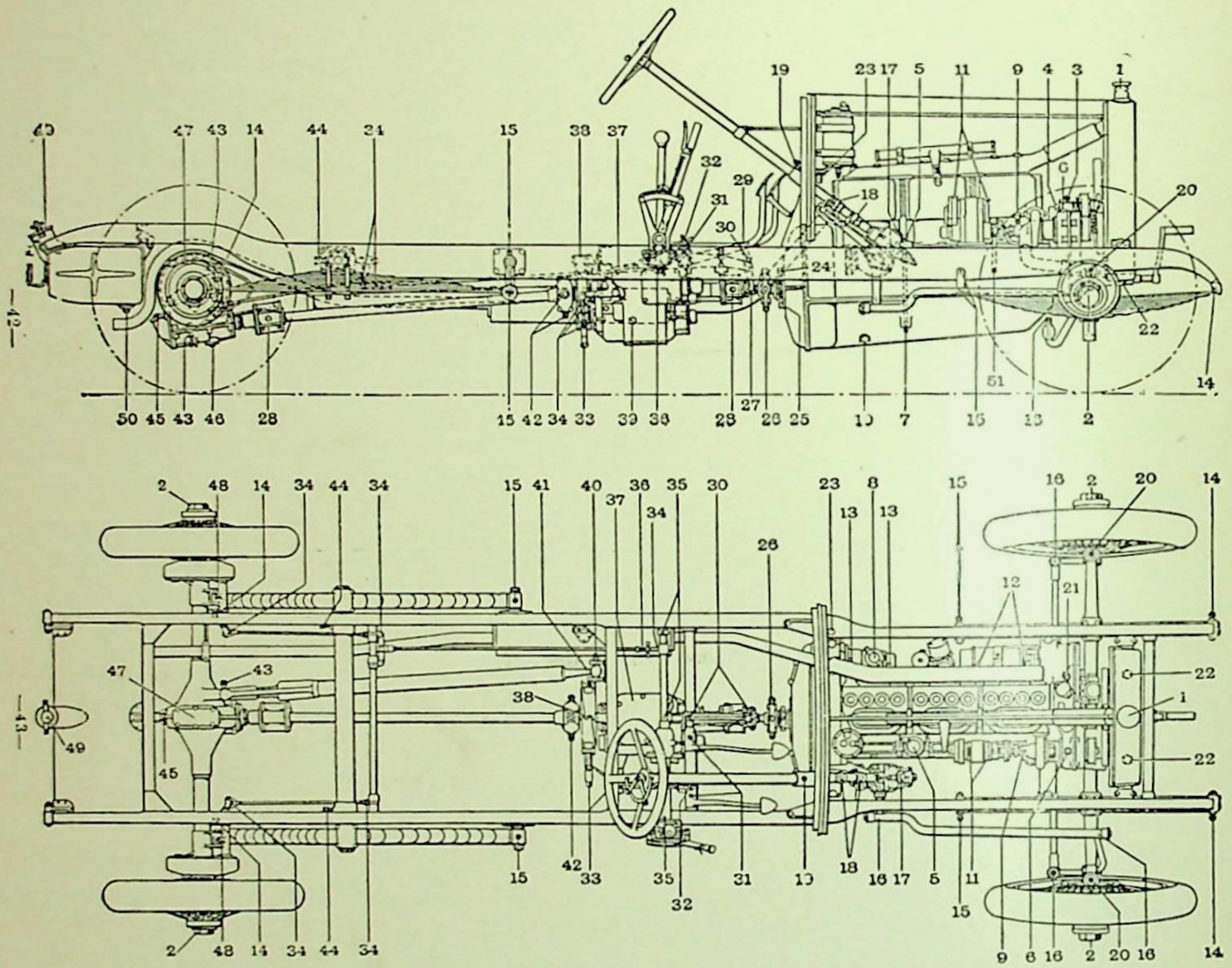


Fig. 289.—Illustrating the Wolseley 20 h.p. Chassis Lubrication.

1. Radiator filling cap (fill through strainer).
2. Grease the hub bearings (remove the hub cap).
3. Oil the fan bearing.
4. Adjustment for fan belt.
5. Crank chamber oil filler.
6. Chain case oil plug.
7. Oil level overflow valve.
8. Engine oil strainer.
9. Grease the pump spindle.
10. Oil reservoir drain plug.
11. Oil the magneto.
12. Oil the dynamo.
13. Oil the starter.
14. Grease the spring pins.
15. Grease the spring shackle pins.
16. Grease the steering joints.
17. Steering box oil plug.
18. Oil the control blocks.
19. Grease the steering column bearing.
20. Oil the swivel pins.
21. Oil drain for tyre pump air filter.
22. Drain for cooling system.
23. Petrol cock and filter.
24. Oil plug for clutch.
25. Drain plug for clutch.
26. Oil the universal pins.
27. Oil the thrust bearing.
28. Oil the sliding couplings.
29. Clutch pedal adjustment.
30. Oil the clutch lever joint pins.
31. Oil the clutch and brake pedals.
32. Brake pedal adjustment.
33. Foot brake adjustment.
34. Oil the brake rod joints.
35. Oil the change speed sliding tube.
36. Side brake adjustment.
37. Replenish oil in gear box.
38. Oil the foot brake bearings.
39. Gear box oil level plug.
40. Grease the torque bar suspension pin.
41. Oil the torque bar ball socket.
42. Oil the universal joints.
43. Grease the torque bar swivel pins.
44. Grease the spring cross tube bearings.
45. Axle case oil filler.
46. Axle case oil drain plug.
47. Axle case inspection cover.
48. Grease the wheel brake spindles.
49. Petrol tank filler (fill through strainer).
50. Petrol tank drain.
51. Carburettor jacket drain.

As regards the steering gear, an account of the lubrication of the various components is given towards the end of Chapter VI. of Vol. II.

In the case of the laminated springs, it is mentioned later that these should be periodically lubricated with a graphite, or thick grease, if of the exposed type, and if enclosed in leather gaiters or puttees, with the lubricating oil recommended for the purpose.

The shackle pins, an example of which is shown in Fig. 24, Vol. II, should be provided either with screw-down greasers or dust-proof oil cups and lubricated at frequent intervals.

The front-axle bearings are usually packed with a medium consistency grease, such as Price's Belmoline, once every season, and the felt retaining washer at the pivot end should be renewed as soon as it shows signs of allowing the grease to leak out.

Amongst the most important parts, from the lubrication viewpoint, is the steering pivot pins, which are constantly in use; any lack of lubrication is at once reflected in a harder steering effort. Screw-down grease caps are generally provided, and a turn should be given every time the car is used for more than a few miles, say once daily. The pivot-pin grease holes should be kept clear, otherwise the grease will leak out past the threads of the screw cap.

In many cases the front axle should be jacked up before the greasers are screwed down, since the weight of the car on the pivot-pin thrust bearings (which are the important parts requiring lubrication) usually prevents the flow of the grease, if the thrust bearings are of the plain type. This can easily be detected by the grease emerging from the top end bearing of the pivot pin instead of the bottom one, when the cap is screwed down, in the usual design in which the front-axle pivot bearings are above and below the stub-axle pivot bearing.

The gear-wheel lever and brake-lever bearings require occasional attention, as also do the various joints and fairleads in the brake mechanism.

Most manufacturers usually issue a lubrication chart in their instruction manuals, and recommend this chart to be pasted on a card, or plywood board, varnished, and hung up in the garage.

The actual lubrication instructions will vary, of course, with the design of the car, but it is possible to construct a general diagram of lubrication instructions applicable to most makes of car, since the general design features are common.

Fig. 289 illustrates the lubrication chart of the 20 h.p. Wolseley chassis, in both plan and side elevation, the key to the lettering being given in the text below.

As an example of a light car chassis lubrication chart, Fig. 290, which refers to the 8 h.p. Rover car, is also included; the system adopted to indicate the periods, or rather the running intervals at which lubrication requires attention, will be evident from the diagram.

In the case of a few highly developed designs of chassis, the lubrication of the principal chassis parts, such as the spring shackle pins, the universal and propeller shaft bearings, the pivot-pin and steering-box bearings, etc., is automatic, an oil pump being fitted which supplies oil through a series of copper pipes (after filtering the oil) to the various parts requiring lubrication.

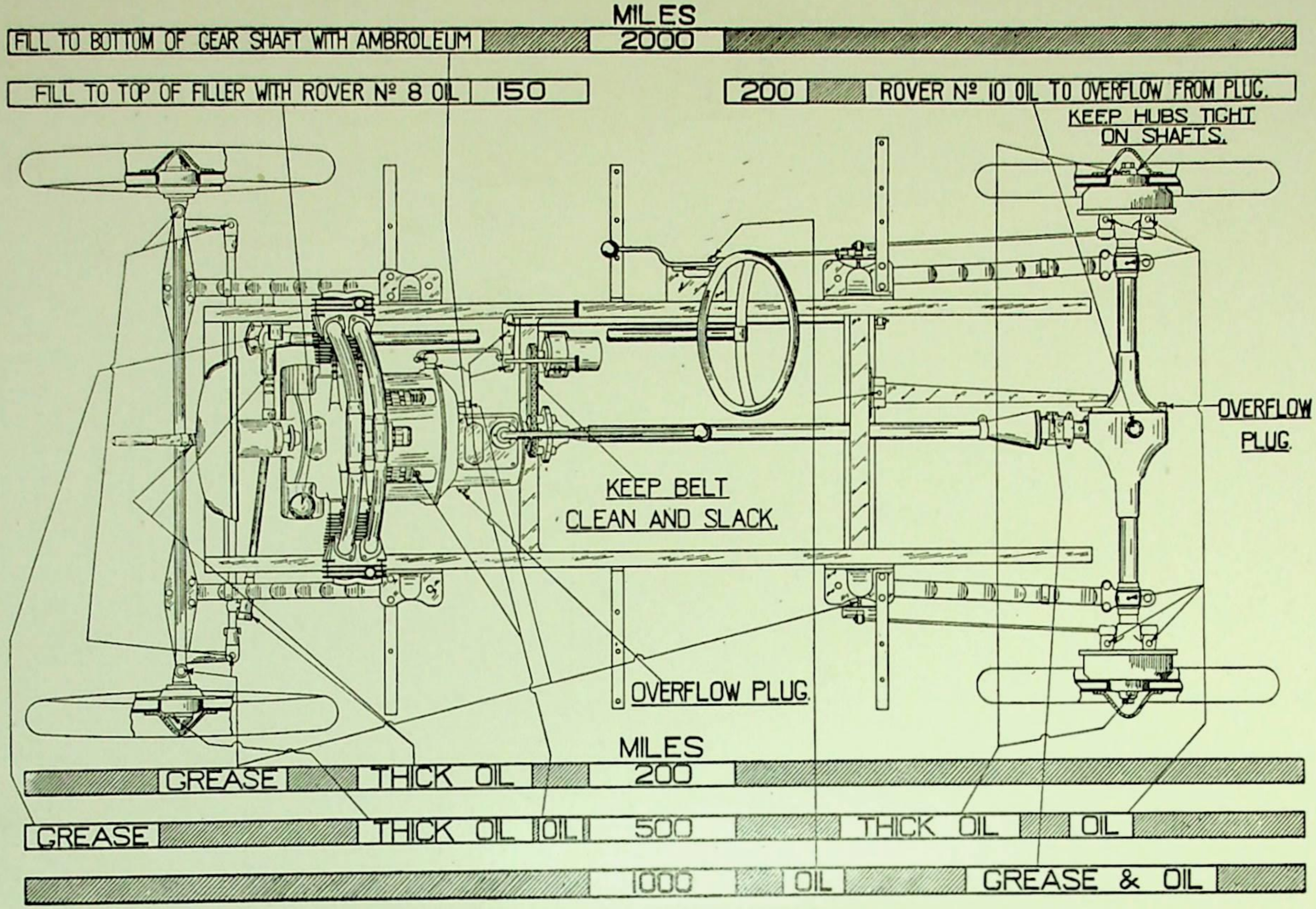

MILES
FILL TO BOTTOM OF GEAR SHAFT WITH AMBROLEUM
2000
FILL TO TOP OF FILLER WITH ROVER Nº 8 OIL
150
200
ROVER Nº 10 OIL TO OVERFLOW FROM PLUG.
KEEP HUBS TIGHT ON SHAFTS.
OVERFLOW PLUG.
KEEP BELT CLEAN AND SLACK.
OVERFLOW PLUG.
MILES
GREASE
THICK OIL
200
GREASE
THICK OIL
OIL
500
THICK OIL
OIL
1000
OIL
GREASE & OIL

Fig. 290.—The 8 h.p. Rover Lubrication Chart.

In the case of the Leyland Eight Car, the movements of the rear single cantilever springs work a plunger in an oil-fed cylinder acting as a buffer stop, and forces oil to all working parts of the chassis. Thus the oiling is dependent upon the movement of the car on the road ; bad roads will therefore produce more lubrication than very smooth roads.

Fig. 291 shows the principle of the Kemp system of chassis lubrication, in which the movement of the rear axle, under the influence of the springing, works the plunger of an oil pump, situated in the side of an oil reservoir, and thus forces oil through a non-return valve to a series of pipes leading to the bearings of the springs. The upward movement of the axle works the plunger upwards by virtue of the inside spring, and the downward movement forces it down due to the pull of the cable and outside spring. In order that the heavy springing shocks shall not interfere with the working of the pump, the stiff coil spring is introduced, and this remains closed under normal circumstances.

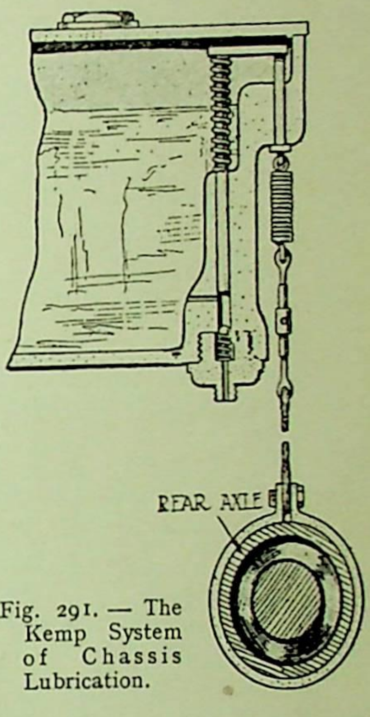


Fig. 291. — The Kemp System of Chassis Lubrication.

END OF VOL. I

Made and Printed in Great Britain by
Hazell, Watson & Viney, Ld., London and Aylesbury.

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3 0402 00298 9012